DESIGN OF A WASTE HEAT EXCHANGER

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ABSTRACT

DESIGN OF A WASTE HEAT EXCHANGER

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The material presented in this Major Technical Report covers the detail design of a heat exchanger used for recovery of waste heat energy from process exhaust flue gases.

Thermodynamic and fluid flow calculations sizing the waste heat exchanger are presented followed by an assessment of flow induced tube vibrations using current literature on the State of the Art.

Detail design of the pressure parts and supports to the ASME Boiler and Pressure Vessel Code Section VIII, Division 1 is included. The design and analysis of Ancillary Piping to the Refinery Piping Code ANSI B 31.3 is also presented. Details of other auxiliary equipment such as the steam drum, gas inlet and outlet cones are given in Appendices A and B.

A complete set of engineering drawings for the entire waste heat exchanger system is included in Appendix C.

ACKNOWLEDGEMENTS

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LIST OF SYMBOLS

Unless otherwise defined in the text, the list of the symbols used are as follows:

CHAPTER 2

a	Constant
a_F	Free flow area
A	Heat transfer surface area
A_{REQ}	Required heat transfer surface area
B	Length of flow area = support plate spacing
C'	Clearance between tubes
C_p	Specific heat
D	Diameter
D_s	Shell diameter
d_i	Internal diameter
D_e	Hydraulic diameter
f	Friction factor
G	Mass flow rate per unit area based on pipe internal diameter
H	Available head for circulation
h_f	Enthalpy of feedwater
h_s	Enthalpy of steam leaving the drum
h_F	Gas film coefficient
ID	Inside diameter of shell
K	Thermal conductivity of gas at bulk gas temperature
K_s	Thermal conductivity of steel
L	Pipe length
L_T	Tube length

L_1	Height from point of average fluid conditions to top tubesheet
L_2	Height to normal water level from top tubesheet.
$\dot{M}_G$	Mass flow rate of flue gas
$\dot{M}_S$	Mass flow rate of steam.
N	Number of support plates
P	Pressure
P_T	Tube pitch
ΔP	Pressure drop
$\dot{Q}$	Heat energy flow rate
$\dot{Q}_S$	Rate of heat energy absorbed by steam
$\dot{Q}/A$	Heat flux
R	Recirculation rate
Re	Reynolds number
r	Latent heat absorption by steam
S	Specific gravity
T_1	Gas inlet temperature
T_2	Gas outlet temperature
T_g	Bulk gas temperature
T_S	Tube wall temperature
t	Tube thickness
ΔT	Arithmetic temperature difference
ΔT_{LM}	Logarithmic temperature difference
Δt_W	Difference between tube wall temperature and saturation temperatures
U_C	Gas film heat transfer coefficient inside tubes
U	Overall heat transfer coefficient
V	Mean specific volume

V_f	Specific volume of liquid
V_g	Specific volume of vapour
V_m	Specific volume of mixture
θ_1	Temperature difference between gas and water - steam mixture at gas inlet
θ_2	Temperature difference between gas and water at gas outlet
ρ	Density
μ_w	Viscosity at tube wall temperature
μ	Mean absolute viscosity
ϕ_s	The viscosity ratio $(\mu/\mu_w)^{0.14}$

CHAPTER 3

a_s	Bundle cross flow area
A	Free flow area parallel to tubes
A_M	Tube cross sectional metal area
A_T	Tube external projected area
B	Minimum support spacing
B_t	Support plate thickness
C	Mode constants
C'	Clearance between tubes
C_d	Viscous damping coefficient
C_L	Fluctuating lift coefficient of vortex
C_n	Normalised damping coefficient
C_T	Minimum clearance between tubes
d	Tube diameter
D	Tube outside diameter
E	Modulus of elasticity
f	Frequency of forcing function
F_B	Tube support plate clearance factor
f_N	Natural frequency of tube
g_c	Gravitational constant
I	Moment of inertia
ID	Shell internal diameter
Hz	Hertz
K	Magnification factor
K_c	Connors number
K_vV	Maximum cross flow velocity

ℓ	Tube span length
L	Longitudinal spacing between tubes
M	Tube mass per unit length
M_F	Added mass coefficient factor
$\dot{M}_S$	Mass flow rate of water-steam
N_{BD}	Baffle type damage number
N_{CD}	Collision type damage number
P_T	Tube pitch
R	Recirculation rate
R_e	Reynolds number
S_M	Maximum allowable fatigue stress
S_t	Strouhal number
T	Transverse spacing between tubes
U	Flow velocity
U_{ACT}	Actual flow velocity
U_{CRIT}	Critical flow velocity
U_P	Velocity parallel to tubes
V	Specific volume of water-steam
W	Total weight per foot run of tube
W_t	Weight of empty tube
W_{fi}	Weight of fluid inside tube
W_{fo}	Weight of fluid displaced by tube
X	Maximum dynamic deflection
X_L	Longitudinal spacing ratio of tubes
X_t	Transverse spacing ratio of tubes
δ	Logarithmic decrement
ρ	Density of water-steam

ζ	Damping factor
ω	Angular velocity of vibration
ω_n	Angular velocity of vibration at resonance

CHAPTER 4

a_p	Plate width
A_r	Areas for reinforcement
A	Code factor for external factor calculations
A_c	Seismic acceleration factor
A_p	WHE projected area
b	Gasket width factor
B	Allowable stress for external pressure
b_p	Plate width
b_o	Half gasket width
C_c	Wind exposure factor
C_g	Wind gust factor
C_p	External pressure coefficient
C'	Corrosion allowance
C	Flat cover design factor
C_n	Cross section or roughness coefficient
d	Diameter of opening
d_c	Manway cover diameter
D	Diameter
d_o	WHE outside shell diameter including insulation
E	Joint efficiency
F_M	Ring force
F_N	Thermal friction force
F_W	Wind shear force
F_Y	External axial force applied to flange joint
F_V	Shear load

f_1	Ring force
G	Gasket mean diameter
h	Height of WHE subject to wind pressure
h_g	Radial distance from gasket load reaction to the bolt circle
I_B	Moment of inertia of lower ring
I_T	Moment of inertia of upper ring
K_1	Support ring force factor
K_2	Support ring moment factor
L_T	Total tube length
m	Gasket factor
M_E	External moment applied to flanged joint
M_r	Ring moment
M	Moment
N	Gasket width
P_L	Load
P_{FD}	Total operating flange design pressure
P	Design pressure
P_a	Allowable external pressure
P_{EQ}	Equivalent pressure to allow for external moment on flange joint
P_W	Wind design external pressure
q	Wind reference velocity pressure
R	Inside radius of shell
r_c	Radius to neutral axis of support
R_A, R_B, R_C	Reactions
R_M	Ring forces
S	Code allowable stress

S_H	Flange Hub stress
S_R	Flange radial stress
S_T	Flange tangential stress
S_p	Horizontal seismic shear force
t	Minimum thickness
t_c	Cover thickness
t_{rn}	Required nozzle neck thickness
V_{PT}	Earthquake lateral shear force
V_W	Wind shear per bracket
W_{ATM}	Bolt loading
W_{m_1}	Required bolt load for operating condition
W_{m_2}	Gasket seating load
W_p	Weight of waste heat exchanger
Y	Gasket seating stress
y	Distance from neutral axis to point under consideration
τ	Shear stress.

CHAPTER 5

A_t Tube cross sectional metal area

D Tube diameter

E Modulus of elasticity

E* Effective modulus of elasticity for perforated tube plate

G Gravitational constant

h Nominal width of ligament at the minimum cross section

I_{zz} Moment of inertia of tube

N Number of tubes

P Tube pitch

P_D Tube pressure

P_S Tube force induced by shell side pressure

P_T Tube force induced by tube side pressure

ROT_Z Rotation about Z axes

S_M Code allowable design stress intensity

t Thickness of tube plate

U_x Displacement in x-direction

U_y Displacement in y-direction

x Coordinates on x-axis

y Coordinates on y-axis

ΔT Temperature of tubes above the shell temperature

ν Poissons Ratio

ν^* Effective Poisson's ratio for perforated plate

α Coefficient of thermal expansion

σ Stress

CHAPTER 6

B_W	Butt welding
C	Corrosion allowance
D_o	Outside diameter of pipe
D	Nominal pipe size
E	Joint efficiency
E_a	Modulus of elasticity of piping material
f	Fatigue factor
g	Gravitational constant
L	Developed length between anchors
L_R	Long radius
P	Internal design pressure
S	Allowable stress
S_c	Allowable stress for material at minimum metal temperature expected during displacement cycle
S_h	Allowable stress for material at maximum metal temperature expected during displacement cycle
S_A	Allowable stress range
STD	Standard
t	Pressure design thickness
U	Anchor distance, straight line distance between anchors
W	Total weight of pipe per unit length
Y	Resultant of total displacement strains to be absorbed by the piping system
y	Material coefficient
ΔT	Increase in temperature of pipe above stress free temperature
ΔX_N	Expansion along x-axis during normal operation

ΔX_S	Expansion along x-axis during start up
ΔY_N	Expansion along y-axis during normal operation
ΔY_S	Expansion along y-axis during start up
α	Coefficient of thermal expansion

APPENDICES

A_e	Effective area of reinforcement due to excess metal thickness
A_{REQD}	Required area of reinforcement
B	Inside diameter of flange
C	Corrosion allowance
D	Inside diameter of cone
E	Joint efficiency
E_R	Modulus of elasticity of reinforcing material
E_S	Modulus of elasticity of shell
g_o	Thickness of hub at small end
K	$\frac{S_s\ E_s}{S_R\ E_R}$
M_o	Total moment acting upon flange
P	Design pressure
R	Inside radius of shell
R_L	Inside radius of large cylinder at junction
R_S	Inside radius of small cylinder at junction
S	Code allowable stress value
S_A	Allowable stress for combined loads.
S_R	Allowable stress for reinforcing material
S_s	Allowable stress for shell material
t	Minimum required thickness of flange
t_F	Flange thickness
t_c	Nominal thickness of cone at cone to cylinder junction exclusive of corrosion allowance
t_s	Nominal thickness of cylinder at cone to cylinder junction, exclusive of corrosion allowance
Y	Flange design factor

$\tilde{\Delta}$	Value to indicate need for reinforcement at cone to cylinder intersection having a half apex angle $\alpha \leq 30°$
α	One half of the included (apex) angle of the cone.

CHAPTER 1

CHAPTER 1

INTRODUCTION

This report describes the design of a waste heat exchanger installed downstream from an existing fluid catalytic cracking unit in a Montreal Petroleum Refinery (FCIM). The schematic layout of the equipment in the overall system is shown in Fig. 1.1.

The FCIM unit produces a flue gas flow rate of up to 200,000 lb/hr during continuous operation at temperatures of up to 1275°F. The purpose of the waste heat exchanger is

(1) cool gases to permit gas cleaning as required by the Clean Air Act of 1975,

(2) facilitate recovery of the catalyst, and

(3) the recovery of heat energy in form of steam.

The waste heat exchanger (WHE) will cool the gases to about 460°F from the gas inlet average temperature of about 1250°F and in the process, will generate about 43,200 lb/hr of saturated steam at 155 psig.

The waste heat exchanger and accessories are shown schematically in Fig. 1.2. The flue gases from the FCIM unit enter the WHE through the gas inlet at the top of the unit. The hot gases are cooled by passing over the cooling surface as they flow down the tubes and out through the gas outlet section to the electrostatic precipitator. The feedwater is introduced into the steam drum. The boiler water flows from the steam drum through downcomer pipes into the WHE shell. The water is heated by the flue gases and steam bubbles are formed as it flows outside the tubes and upwards inside the WHE shell. The water and steam mixture exits the WHE shell through riser pipes and flows into the steam drum where the steam is seperated from the water and is available for the refinery process use. The condensate plus make up feedwater is circulated through the WHE system where the heat pick up process is repeated continuously.

The engineering drawings for the various components are shown in Appendix C. The detail design of these components is described in this report.

Thermal design and verification of adequate water circulation through the WHE are covered in Chapter 2 which is followed by an investigation of flow induced tube vibration in Chapter 3.

The detail design of the WHE pressure parts and supports is outlined in Chapter 4. The report continues with design by analysis of the tube plates and tubes supports in Chapter 5.

The penultimate part of the report, Chapter 6, addresses itself to the design and flexibility analysis of the piping associated with the WHE system.

Finally the design of the other major components such as the gas inlet and outlet conical sections and steam drum are presented in the Appendices.

One aim of any analysis is to determine, within reasonable accuracy, values of the dependent variables for the given values of independent variables. It seems largely a matter of style which variables we regard as independent. What we seek is a set of equations which have as a consequence the bounding of whatever quantities we feel are significant.

It is apparent from above that the boundness of our solution is a measure of the thoroughness of the analysis, but this does not imply that the analysis is therefore exact. Our notation of what is exact seems to stem from an intuitive knowledge of a set of well-defined physical phenomena. Reduction of a complex phenomenon to a set of simpler, well-defined phenomenon is generally agreed to be the chief aim of analysis.

This means that a good analysis has two distinct characteristics:

(i) it relies upon reducibility of complex problems to "irreducible" familiar problems,

(ii) the numerical results must bear good resemblance to the size of corresponding quantities found in actual physical situations.

The design of the waste heat exchanger as presented in this report reflects the significance of the above concepts which demonstrates the importance of good engineering judgement.

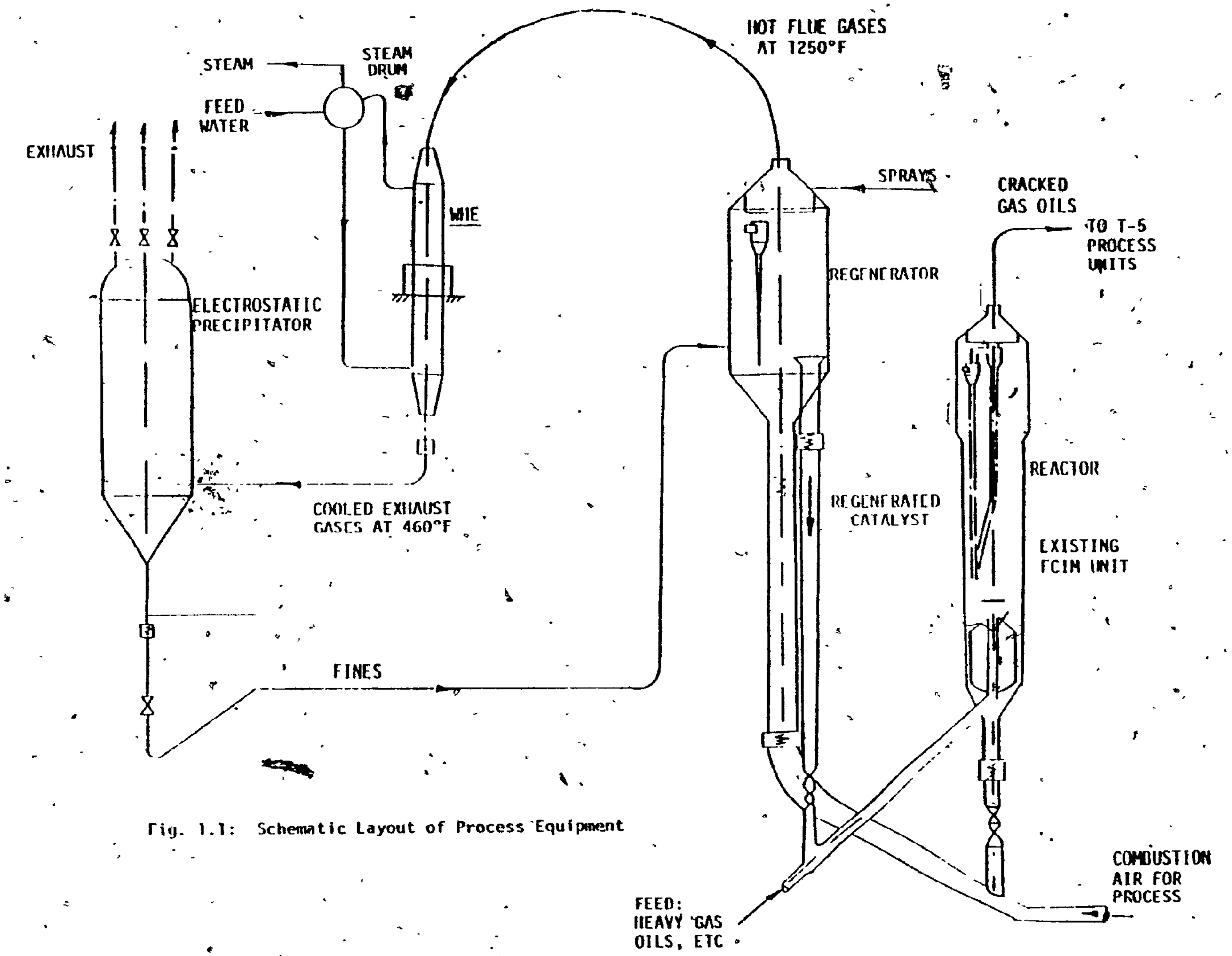


Fig. 1.1: Schematic Layout of Process Equipment

Schematic Outline of Waste Heat Exchanger System

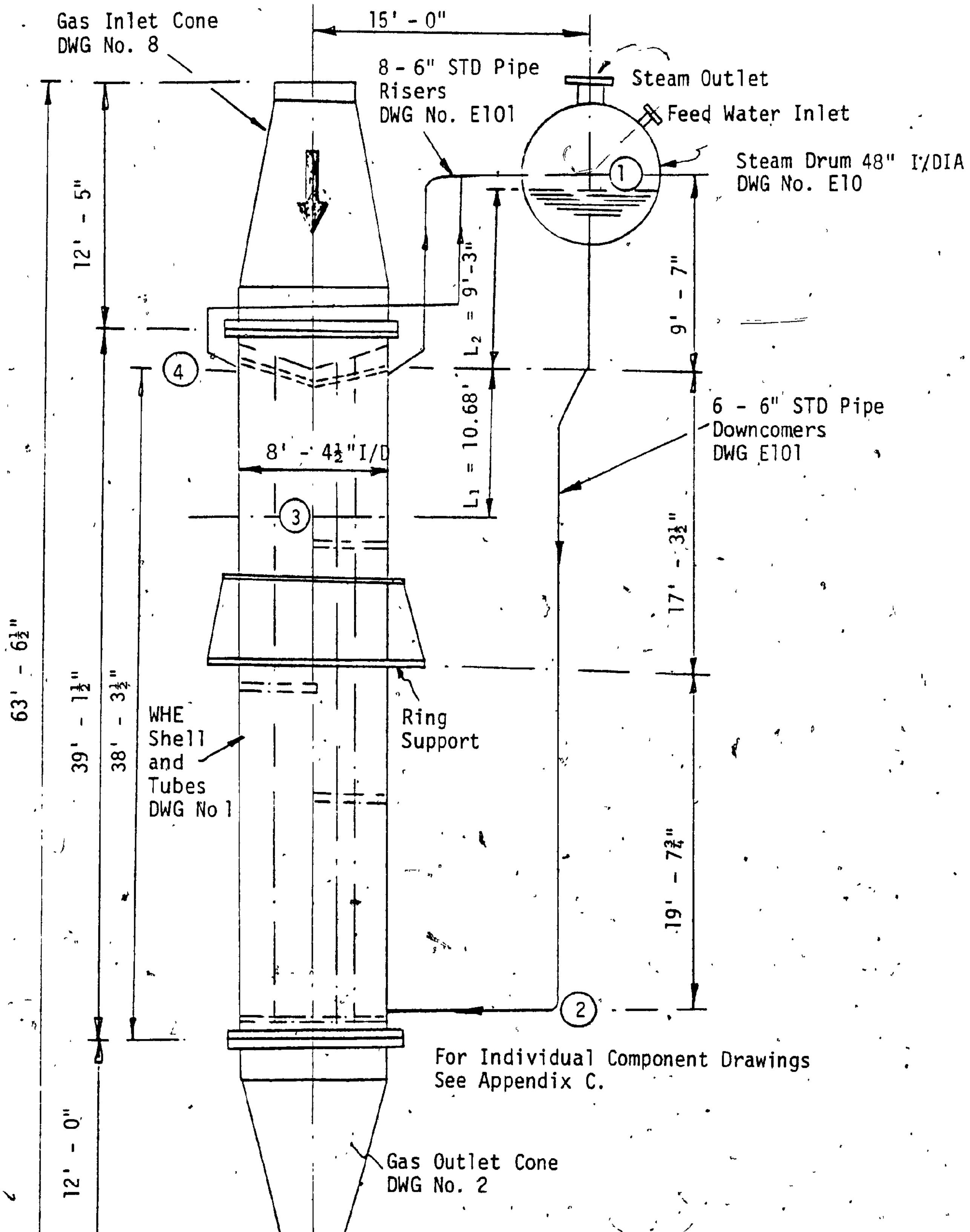


Fig. 1.2

General Fabrication and Performance Information

FIGURE G-5.2

HEAT EXCHANGER SPECIFICATION SHEET

Customer	Montreal Refinery	Job No.	
Address		Reference No.	
Plant Location	FCIM	Proposal No.	
Service of Unit	Regenerator Exhaust Gas WHE	Date Jan.80 Rev.	
Size 8'-6"dia x 37'-6" Type (Hor/Vert)		Item No.	
Surf/Unit (Gross/Eff) 9118 Sq Ft.	Shells/Unit 1	Connected In Parallel Series	Surf/Shell (Gross/Eff) Sq Ft

PERFORMANCE OF ONE UNIT

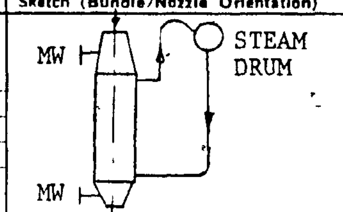
	Shell Side		Tube Side	
Fluid Allocation				
Fluid Name	Water and Steam		Exhaust Gases	
Fluid Quantity, Total Lb/Hr			186 395 (+8333 Catalyst)	
Vapor (In/Out)				
Liquid	43 200			
Steam	—	43 200 (2)		
Water				
Noncondensable			186 395 +	8333
Temperature (In Out) °F	230	368	1250	450
Specific Gravity				
Viscosity, Liquid Cp				
Molecular Weight, Vapor				
Molecular Weight, Noncondensable			29.58	
Specific Heat at 850°F Btu/Lb °F	1		0.2773 Gas	0.262 Catalyst
Thermal Conductivity Btu Ft/Hr Sq Ft °F				
Latent Heat Btu/Lb @ °F				
Inlet Pressure in drum Psig	155		15.3	
Velocity Ft/S			100 at Tube Inlet	
Pressure Drop, Allow./Calc. Psi	/		/	
Fouling Resistance (Min.)	0.002		0.004	

Heat Exchanged 43 092 000	Btu/Hr; MTD (Corrected)		°F
Transfer Rate, Service 14.23	Clean 15.71		Btu/Hr Sq Ft °F

CONSTRUCTION OF ONE SHELL

	Shell Side	Tube Side
Design/Test Pressure Psig	180 / 270	40 / None
Design Temperature °F	650	650 / 1400
No. Passes per Shell	1	1
Corrosion Allowance In.	1/8	—
Connections Size & Rating — In	R.F.	R.F.
Out	R.F.	R.F.
Intermediate		

Sketch (Bundle/Nozzle Orientation)



Tube No. 470 OD 2.5 In.; Thk (Min/Avg) 0.203 In.; Length 36.25 Ft; Pitch 3.5 In. △ 60			
Tube Type	Seamless	Material	C.S. SA 192
Shell	ID 100.5 OD 102 In	Shell Cover	— (Integ.) (Remov.)
Channel or Bonnet	—	Channel Cover	C.S. Refractory Lined (3)
Tubesheet-Stationary	C.S. SA 516 GR 70	Tubesheet-Floating	—
Floating-Head Cover	—	Impingement Protection	S.S. Ferrules
Baffles-Cross	Type	% Cut (Diam/Area)	Spacing: c/c Inlet In.
Baffles-Long		Seal Type	
Supports-Tube	Three C.S. U-Bend	Type	
Bypass Seal Arrangement	—	Tube-Tubesheet Joint	—
Expansion Joint	—	Type	—
ρv²-Inlet Nozzle	— Bundle Entrance —	Bundle Exit	—
Gaskets-Shell Side	S.S. Jacketed Asbestos	Tube Side	S.S. 410 Jacketed Asbestos
-Floating Head	—		
Code Requirements	ASME Section VIII Div. 1	TEMA Class	R
Weight/Shell	Filled with Water 321 000	Bundle	Lb

Remarks

(1) Vertical Arrangement Thermosyphon WHE, Gas Flow Down.
(2) Saturated Steam From Steam Drum.
(3) Refractory Lined Gas Inlet and Outlet Cones.
(4) Conical Tubesheet to Avoid Vapour Blanketting.
(5) Tubeside Hydrotest was Replaced by Full Radiography of all Butt Welds.

TABLE 1.1

CHAPTER 2

CHAPTER 2

THERMAL DESIGN AND VERIFICATION OF ADEQUATE WATER CIRCULATION

2.1 HEAT TRANSFER CALCULATIONS

The basic design information is provided on the data sheet shown in Table 1.1. Standard data sheets like this are used to convey customer's requirements for proposed heat exchangers to fabricators. This data sheet specifies all process conditions, fouling factors and basic data for the thermal and mechanical design of the unit.

2.1.1 Heat Balance

Heat energy available from flue gases and the catalyst which is entrained in the gas stream is calculated using the following equation:

$$\dot{Q} = \dot{M}\, C_p\, (T_1 - T_2)$$

Heat energy from flue gas = 186 395 x 0.2773(1250 - 450)

= 41 349 866 Btu/hr.

Heat energy from catalyst = 8 333 x 0.262(1250 - 450)

= 1 746 600 Btu/hr.

$\dot{q}_{Total}$ = 43 096 466 Btu/hr.

Water-steam

$$\dot{Q}_s = \dot{M}_s (h_s - h_f)$$

$$= 43\,200(1195.5 - 198) = 43\,092\,000 \text{ Btu/hr.}$$

2.1.2 Heat Transfer Coefficients

Reference [1]* "Heat Transfer and Draught Loss in the Tube Banks of Shell Boilers", gives the following equation for heat transfer inside tubes:

$$U_c = \frac{aK}{D} Re^{0.8} \quad \text{Btu/hr ft}^2{}^\circ\text{F}$$

where

a = constant = 0.022

D = inside tube diameter = $\frac{2.5 - 0.44}{12}$ = 0.1717 ft

T_g = bulk gas temperature

$$= \frac{T_s + (T_1 - T_s) - (T_2 - T_s)}{\log_e \frac{T_1 - T_s}{T_2 - T_s}}$$

T_s = Tube wall temperature = 368 + 18 = 386°F

T_1 = 1250°F $\qquad T_2$ = 450°F

$$T_g = \frac{386 + (1250 - 386) - (450 - 386)}{\log_e \frac{1250 - 386}{450 - 386}} = 456^\circ\text{F}$$

K = thermal conductivity of gas at bulk gas temperature T_g, $\frac{\text{Btu}}{\text{ft sec}^\circ\text{F}}$

$$= 0.0215 \frac{\text{Btu}}{\text{ft hr}^\circ\text{F}}$$

$$= 5.972 \times 10^{-6} \frac{\text{Btu}}{\text{ft sec}^\circ\text{F}}$$

Re = Reynolds Number = $\frac{DG}{\mu}$

G = Mass velocity of gas, $\frac{\text{lb}}{\text{sec ft}^2}$

*Number in brackets refer to list of references given at the end of the report.

With 470 - 2.5 in. outside diameter tubes:

$$\text{Gas flow area} = 470 \times \frac{\pi}{4} \times 0.1717^2 = 10.882 \text{ ft}^2$$

$$G = \frac{194\,728}{3600 \times 10.882} = 4.9703 \text{ lb/sec ft}^2$$

μ = absolute viscosity at bulk gas temperature

$= 0.062$ lb/hr ft $= 1.7222 \times 10^{-5}$ lb/sec ft

$$R_e = \frac{DG}{\mu} = \frac{0.1717 \times 4.9703}{1.7222 \times 10^{-5}} = 4.955 \times 10^4$$

$$U_c = \frac{0.022 \times 5.972 \times 10^{-6} \times (4.955 \times 10^4)^{0.8}}{0.1717}$$

$= 0.7652 \times 10^{-5} \times 5702.32$

$= 0.004364$ Btu/sec ft^2°F

$= 15.71$ Btu/Hr ft^2°F

With gas flow under pressure the heat transfer will be enhanced by about 4 to 5%. Reference [1] does not take into account any fouling factors. Conservatively we can allow for these and take U_c to be gas side film coefficient. Then the resistances are as follows:

Water side fouling resistance = 0.002

Gas side fouling resistance = 0.004

Metal resistance $= \frac{t}{K_s} = \frac{0.22}{12} \times \frac{1}{28.6} = 0.000641$

Gas side film resistance $= \frac{1}{15.71} = 0.06365$

Total resistance = 0.07029

Overall heat transfer coefficient including fouling and tube metal resistance = U

$$U = \frac{1}{0.07029} = 14.23 \text{ Btu/hr ft}^2\text{°F}$$

2.1.3 Required Heating Surface

$$\dot{Q} = U.A\ \Delta T_{LM}\ .$$

Logarithmic temperature difference = ΔT_{LM}

$$\Delta T_{LM} = \frac{\theta_1 - \theta_2}{\text{Log}_e \frac{\theta_1}{\theta_2}} = \frac{(1250 - 368) - (450 - 368)}{\text{Log}_e \left(\frac{1250 - 368}{450 - 368}\right)}$$

$$= 337°F$$

$$A_{REQUIRED} = \frac{\dot{Q}}{U\ \Delta T_{LM}}$$

$$= \frac{43\ 096\ 466}{14.23 \times 337} = 8989\ ft^2$$

Actual heating surface provided = 9118 ft^2 with 470 - $2\frac{1}{2}$ in °/D x 6 BWG tubes.

2.2 GAS SIDE PRESSURE DROP

The gas side pressure drop through the waste heat exchanger consists of three components as outlined in reference [1].

$$\Delta P = \Delta P_1 - \Delta P_2 + L_T \Delta P_3$$

where

ΔP = total draught loss in w.g.

L_T = tube length in feet

= 37.75 ft including ferrules

ΔP_1 = pressure drop due to contraction at entry to tube bank

= 3 in. WG from Fig. 5, Ref. [1]

ΔP_2 = pressure rise due to reduction in velocity down tube including allowance for the increase in flow area

= 1.1 in WG from Fig. 6, Ref. [1].

ΔP_3 = pressure drop due to friction

= 0.37 in WG /ft run

ΔP = 3 - 1.1 + 37.75 x 0.37

= 15.87 in WG.

This is acceptable for process conditions under consideration.

2.3 ASSESSMENT OF TUBE METAL TEMPERATURE

In Chapter 5 we will require a realistic estimate of the differential temperature between the tube wall and the cooler WHE shell. The tube temperature is required for the mechanical design of the tubeplates and to check that sufficient number of tube support plates are provided to avoid tube buckling due to compressive tube stresses induced by differential thermal expansion between tubes and the shell.

The waste heat exchanger shell between tubeplates, including support rings and stiffneers, will be completely insulated with 4 in. thick insulation. The top tubeplate will be lined with 7 in. of refractory backed with 2 in. of insultation. The air film heat transfer coefficient from the surface of insulation to the ambient air will be about 2 Btu/hr.ft^2 °F. The heating and cooling of water in the WHE shell will always be very gradual, never exceeding a maximum rate of about 50°F per hour. Under these conditions, with good circulation, the WHE shell temperature will very closely approach the saturation temperature of the water (or water-steam mixture) inside the shell.

The maximum heat input at gas inlet temperature of 1250°F with water steam mixture temperature of 368°F is

$$\left(\frac{\dot{Q}}{A}\right)_{max\ inlet} = h_f \Delta T = 16.5 \times (1250 - 368) = 14553\ \frac{Btu}{hr\ ft^2}$$

where $h_f = 16.5 \dfrac{\text{Btu}}{\text{hr ft}^2\ ^\circ\text{F}}$ includes an allowance for the fact that the gases are under pressure.

$$\left(\frac{\dot{Q}}{A}\right)_{\text{average}} = 16.5 \left(\frac{1250 + 450}{2}\right) - 368 = 7953 \ \frac{\text{Btu}}{\text{hr ft}^2}$$

Figures 2.1 and 2.2 are typical boiling curves of water from pools taken from references [3] and [2] respectively. These curves show that Δt_w, the difference between the tube wall and the vapour temperatures for the above two conditions are:

1. At maximum heat flux of $14553 \ \dfrac{\text{Btu}}{\text{hr ft}^2}$

   $\Delta t_w = 25°\text{F}$

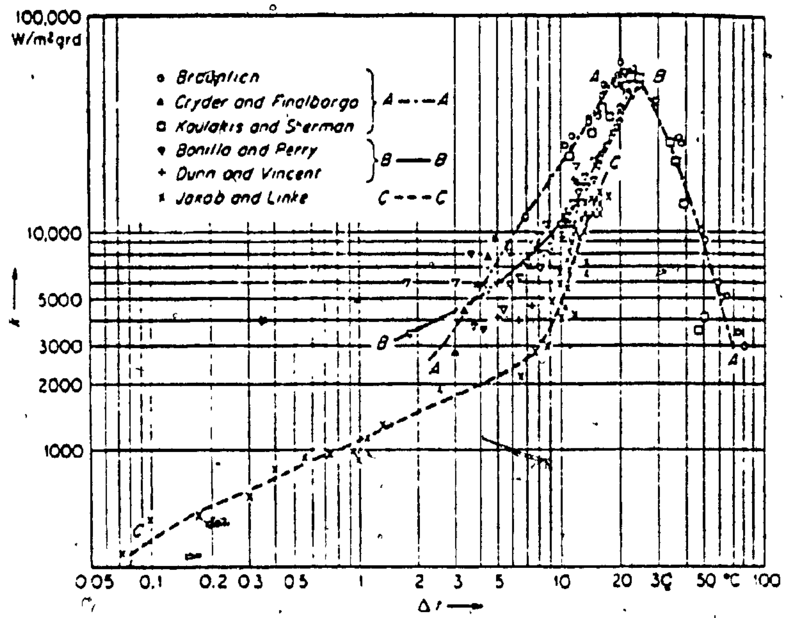
2. At average heat flux of $7953 \ \dfrac{\text{Btu}}{\text{hr ft}^2}$

   $\Delta t_w = 10°\text{F}$

Thus for the specified operating conditon the maximum Δt_w will be about 25°F. The average tube thickness is 0.22 in for 2½"°/D x 6 BWG THK tubes. With the specified fouling factors the temperature drop through the tube wall is about 8°F. These considerations show that the average tube wall temperature will be about 20°F above water temperature. This is also in good agreement with reference [1]. For conservative mechanical design, however, we will take the temperature differential between tubes and shell to be 45°F when calculating tube stresses in Chapter 5.

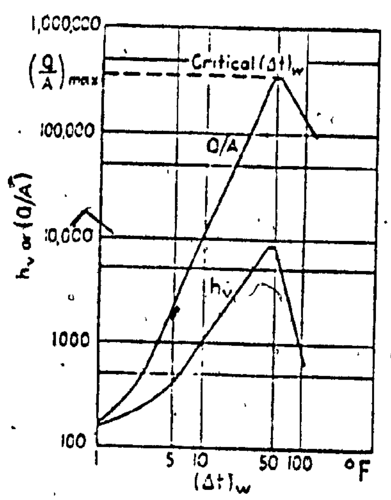
2.4 VERIFICATION OF ADEQUATE WATER CIRCULATION

Feed water is introduced into the WHE system at the steam drum at a temperature of 230°F. In the steam drum 'cold' feed water mixes with steam and saturated water. The resulting mixture in the steam



Heat-transfer coefficient for water boiling on horizontal tubes (A) and on horizontal plates (B, C) at 1 atm. [*From W. H. McAdams, "Heat Transmission," 2d ed. McGraw-Hill Book Company. Used by permission. Copyright 1942.*]

Fig. 2.1



Boiling curve of water from pools. (*After McAdams.*)

Fig. 2.2

drum is at saturation temperature corresponding to the operating pressure in the steam drum. The WHE system flow circuits are shown in Fig. 2.3. The downcomers, which are not heated, connect the steam drum to the bottom of the vertical waste heat exchanger, which acts as heated riser. Figure 2.4 shows the steam water density differential available for natural circulation. In a natural circulation system, the circulation will increase with increased heat input (and increased steam output) until a maximum value is reached, after which further increase in heat asbsorption will result in a decrease in flow. The general form of the curve is shown in Fig. 2.5. Two opposing forces are present. The increase in flow results from the increase in the difference of the densities of the respective fluids in downcomers and risers caused by the increase in heat absorption. However, at the same time the friction and impact pressure losses in both downcomers and risers are increasing. When the rate of increase in these losses caused primarily by the increase in specific volume of steam and water mixture in the riser circuits, becomes greater than the gain from increase in available head due to the density differences, the flow rate will begin to drop until an equilibrium is reached.

A proper objective, therefore, is always to design all the circuits to operate in the region of the rising part of the curve. When the design is limited to the rising portion of the circulation curve, a natural circulation boiler tends to be self-compensating for the numerous variations in heat-absorption surface cleanliness, nonuniform heating conditions and even the inability to forecast precisely the actual conditions over the operating lifetime.

Definitions for Circulation Calculations

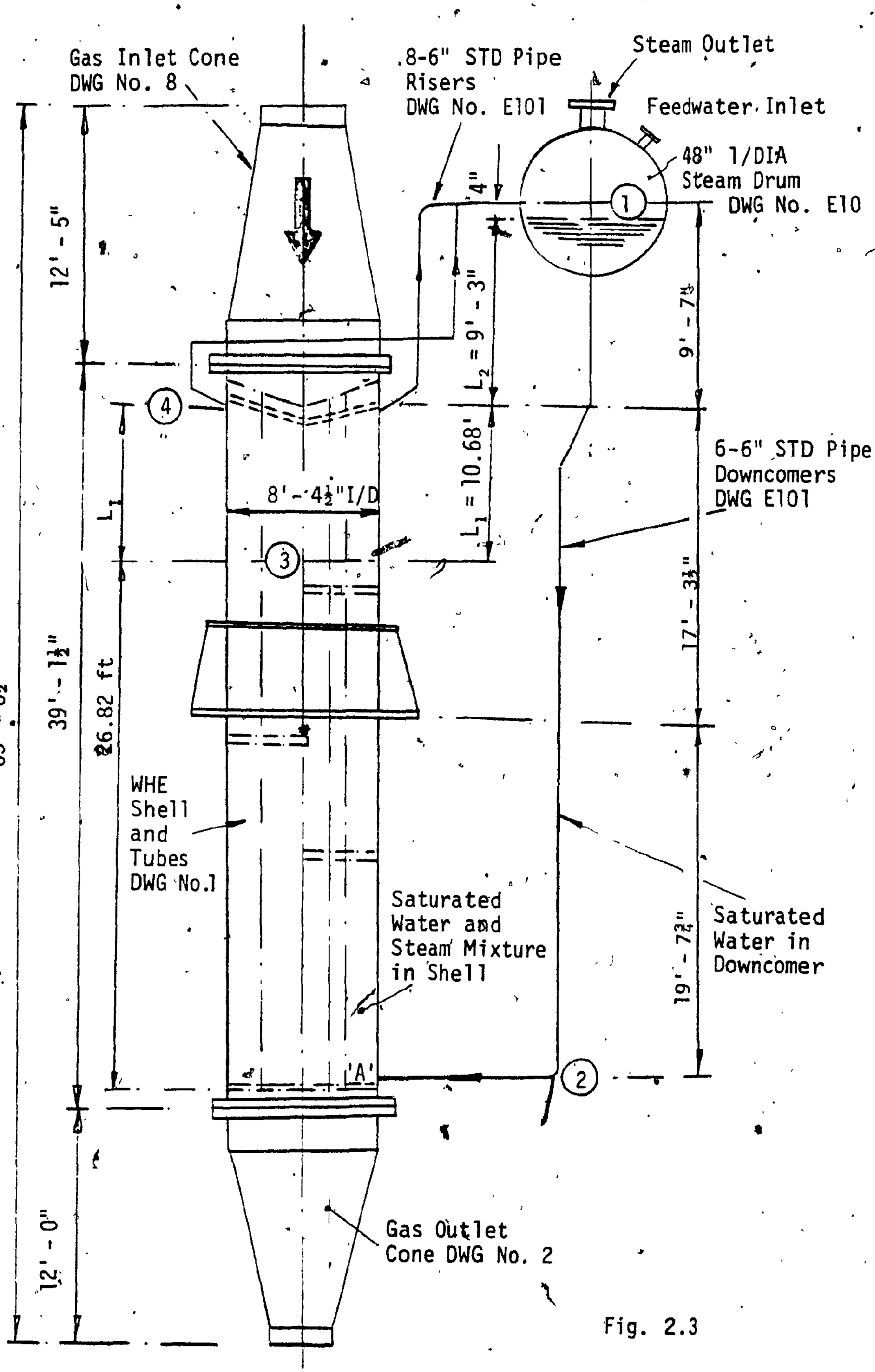
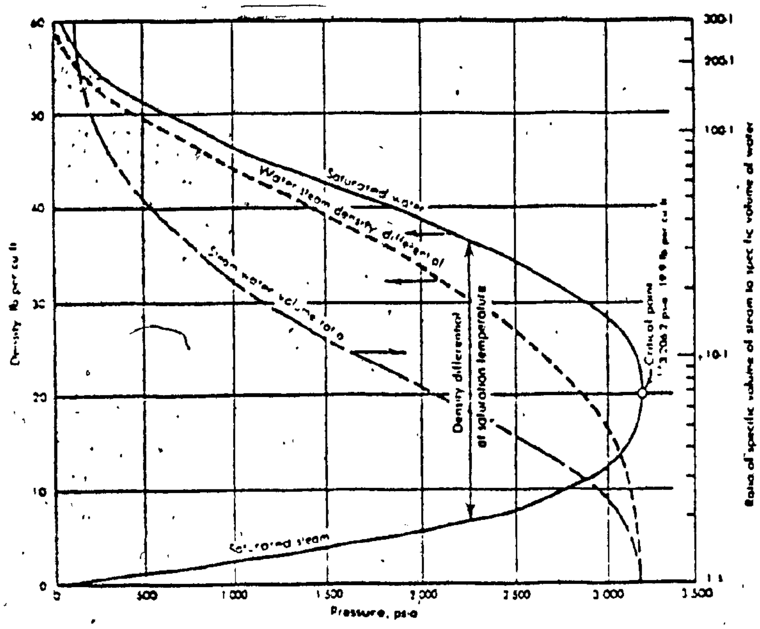
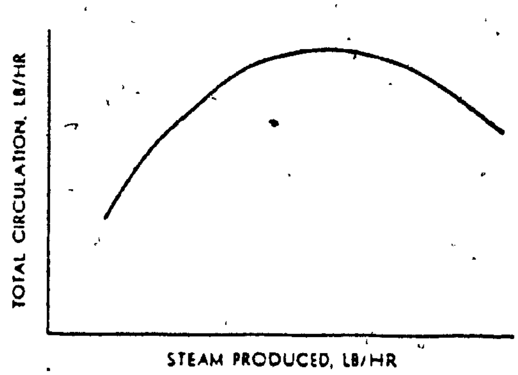


Fig. 2.3



Steam-water density differential available for natural circulation
(From J. H. Keenan and F. G. Keyes, Thermodynamic Properties of Steam.*)*

Fig. 2.4



Typical relationship between circulation in a boiler circuit (at a given pressure) and amount of steam produced (scale arbitrary)

Fig. 2.5

For the waste heat exchanger circuit, shown in Fig. 2.3, the basic relationships noted will be used to demonstrate the principles of natural circulation, simple in conception but somewhat tedious in application.

For stabilized flow (system in equilibrium), the mass flow in the downcomer must equal the mass flow in the riser. Also, the net pressure at 'A' (Fig. 2.3) of the fluid in the downcomer must be balanced by net pressure of fluid in the riser, that is, the net head, H_d, in the downcomer must equal the net head, H_r in the riser. This is illustrated in Fig. 2.6. The state of equilibrium is represented by the point at the intersection of the curves.

In most practical design applications an approximate verification of adequate cooling water circulation is sufficient. The following calculations are a conservative estimate of the water circulation in the waste heat exchanger system.

2.4.1 Available Head for Circulation

With reference to Fig. 2.3 we can state the available head for water circulation is H

$$H = L_1 (\rho_1 - \rho_3) + L_2 (\rho_1 - \rho_4)$$

where L_1 and L_2 are heights as defined in Fig. 2.3.

ρ_1 to ρ_4 are the densities of water or water steam mixture being circulated at points indicated.

In the above expression equivalent of L_1 below point 3 has been neglected to allow for pressure drop in steam drum and yield conservative analysis.

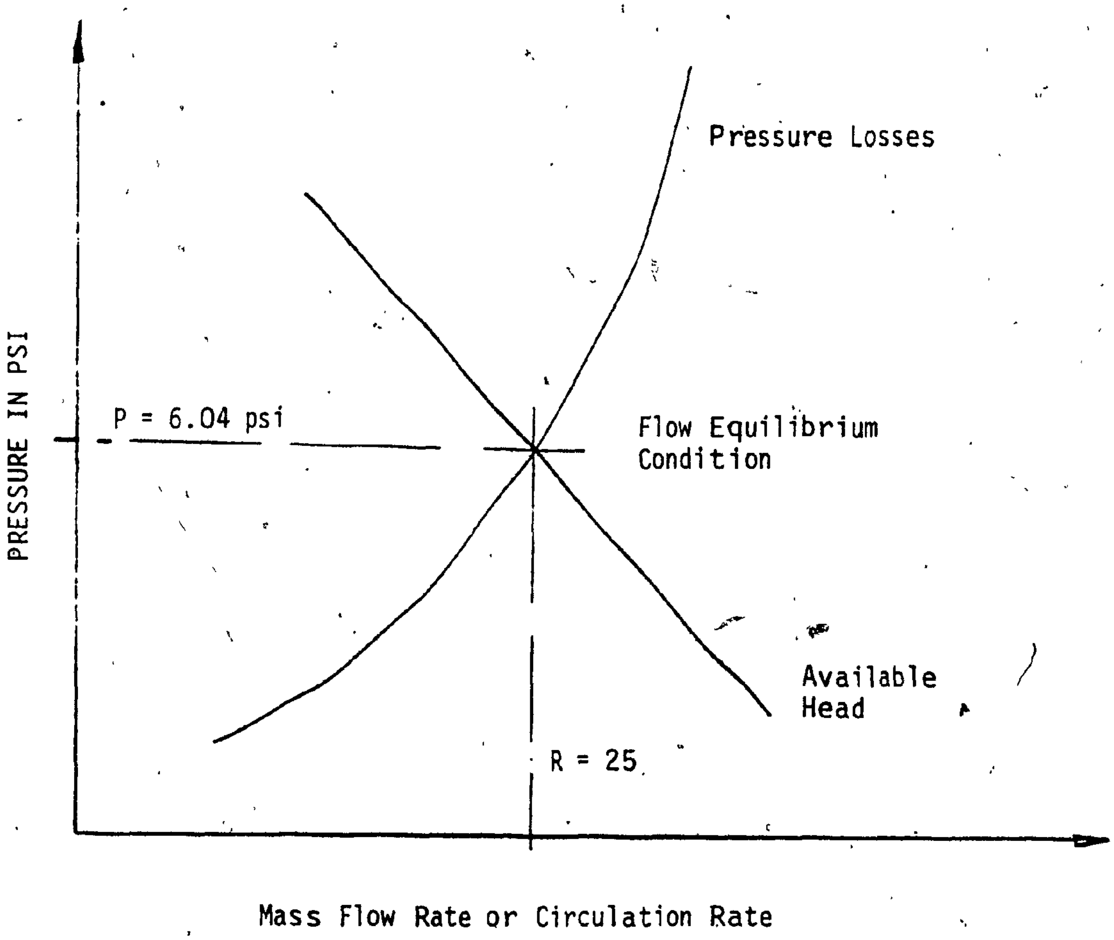


Fig. 2.6: Available Head and Pressure Losses Equilibrium for Specified Waste Heat Exchanger for Constant Output of 43 200 lb/hr.

Under normal operating conditions the pressure at point 1 will be 156 psig. Working from this point we can calculate pressures, specific volumes and densities at all other points. As indicated (see Fig. 2.6) the calculations for the equilibrium flow condition requires a trial and error solution. We will assume a recirculation rate of 25. That is for every pound of steam generated in the WHE there will be an additional 24 lbs of water recirculated through the system. Then we have at point 4:

$$P_4 = P_1 + L_2\,\rho_4$$

$$= 156 + 9.25 \times \frac{8.08}{144}$$

$$= 156.5 \text{ psig} = 171.2 \text{ psia}$$

$$V_f = 0.0182\ \frac{ft^3}{lb} \qquad V_g = 2.657\ \frac{ft^3}{lb}$$

$$V_4 = (1-x)\cdot V_f + xV_g \quad \text{where} \quad x = \frac{1}{R} = \frac{1}{25}$$

$$V_4 = \frac{24}{25} \times 0.0.0182 + \frac{1}{25} \times 2.657 = 0.1237\ \frac{ft^3}{lb}$$

$$\rho_4 = \frac{1}{V_4} = 8.081\ \frac{lb}{ft^3}$$

for point 3

$$V_3 = \frac{1}{2}\ (V_2 + V_4)$$

Properties at various points are tabulated in Table 1.

$$H = L_1(\rho_1 - \rho_3) + L_2(\rho_1 - \rho_4)$$

$$= 10.68\ (54.95 - 14.08) + 9.25\ (54.95 - 8.081)$$

$$= 436.49 + 433.54 = 870.03 \text{ psf}$$

$$H = 6.042 \text{ psi} \quad \text{with} \quad CR = 25$$

TABLE 2.1

PROPERTY \ LOCATION	POINT 1	POINT 2	POINT 3	POINT 4
Gauge pressure psig	156	173.6	—	156.5
Absolute pressure psia	170.7	188.3	—	171.2
V_f specific volume liquid	0.0182	0.0183	—	0.0182
V_g specific volume vapour	2.660	2.430	—	2.657
V_m specific volume ft^3/lb	0.0182	0.0183	0.071	0.1237
P_m specific density lb/ft^3	54.95	65.65	14.08	8.081

Properties of water and steam and water mixture at various locations in the WHE as identified in Fig. 2.3.

2.4.2 Pressure Losses in WHE Circuits

Having calculated the available head with recirculation rate of 25, next step is to calculate the pressure drop through the WHE system with this recirculation rate.

Steam generated in shell = $\dot{M}$.

$$\dot{M} = \frac{\varepsilon \cdot \dot{Q}}{r} = \frac{43\ 096\ 466}{1196.5 - 346.1} = 50\ 678 \text{ lb/hr}$$

with CR = 25 total mass being circulated is 25 x 50 678 = 1 266 950 lb/hr.

The pressure losses in the piping were calculated using the formulae listed below:

1. The loss due to friction ΔP_F is obtained from

$$\Delta P_F = \frac{fL \cdot V}{di} \left(\frac{G}{100\ 000}\right)^2 \quad \text{lb/in}^2$$

where

f = friction factor from reference [4] using Reynolds Number (R_e).

$$R_e = \frac{G\ d_i}{12\ \mu}$$

G = mass flow based on pipe internal diameter $\frac{\text{lb}}{\text{hr ft}^2}$

d_i = internal diameter ---------- in.

μ = mean absolute viscosity --- lb/hr ft

L = pipe length including developed length of bends ----- ft

V = mean specific volume ------ ft³/lb

2. The loss due to bends ΔP_B in addition to the friction loss through the developed length is obtained from

$$\Delta P_B = F_B \frac{V}{12} \left(\frac{G}{100\ 000}\right)^2 \text{ ---------- lb/in}^2$$

where F_B = bend factor

= 0.5 for 90° elbow

= 0.5 for 45° elbow including the increase for close location to another bend.

3. Loss due to inlet and outlet ΔP_S is obtained from

$$\Delta P_S = 1.5 \frac{V}{12} \left(\frac{G}{100\ 000}\right)^2 \text{ ---------- lb/in}^2$$

4. The loss due to a contraction in cross section for configuration used is obtained from

$$\Delta P_C = 0.5\ (1 - RA)\ \frac{V}{12} \left(\frac{G}{100\ 000}\right)^2 \text{ ---------- lb/in}^2$$

where RA = ratio of the smaller area to the larger area.

5. The loss due to an enlargement in cross section for configuration used is obtained from

$$\Delta P_E = 0.8\ (1 - RA)^2\ \frac{V}{12} \left(\frac{G}{100\ 000}\right)^2 \text{ ---------- lb/in}^2$$

The shell-side pressure drop is calculated using the following equation from Kern reference [2]:

$$\Delta P_{Sh} = \frac{f\ G_S{}^2\ D_S\ (N + 1)}{5.22 \times 10^{10} D_e\, S\, \phi_S} \text{ ---------- psi}$$

where G_S was based on $a_S = \frac{ID \times C'\ B}{P_T \times 144}$,

C' = 1 in

B = 7'-6" = 90 in

P_T = 3.5 in

ID = 100.5 in

$D_e = \frac{4 \times \text{free area}}{\text{wetted perimeter}}$

D_S = 8.375 ft

free area = $a_F = \frac{\pi}{4}(8.375)^2 - 470\ \frac{\pi}{4}\left(\frac{2.5}{12}\right)^2 = 39.06\ \text{ft}^2$

wetted perimeter = $470 \times \quad \times \frac{2.5}{12} + 8.375$

= 307.61 + 26.31

= 333.9 ft

$\therefore\ D_e$ = 0.468 ft

$\phi_s = 1$

$S = \frac{\rho_3}{\rho} = \frac{14.244}{62.4} = 0.23$ = specific gravity

$N+1 = 12\ \frac{L_T}{B}'' = \frac{12}{12} \times \frac{37}{7.5} = 5$

$$\Delta P_{Sh} = \frac{0.0017 \times 70819^2 \times 8.375 \times 5}{5.22 \times 10^{10} \times 0.468 \times 0.23 \times 1} = 0.064 \text{ psi Say } \Delta P_{Sh} = 0.15 \text{ psi}$$

The pressure losses in the WHE system are summarized in Table 2.2. From Table 2.2 it can be seen that with circulation rate of 25 available head of 6.042 psi is very close to total pressure losses in the WHE circuits. Hence WHE will have CR of at least 25 thus ensuring adequate cooling water supply to the heating surfaces.

TABLE 2.2

ITEM		PRESSURE DROP PSI
Downcomers: 6 - 6" STD Wall Pipes		
Entry and Exit	ΔP_S	0.337
Friction in Pipes	ΔP_F	0.3442
Bend Loss	ΔP_B	0.2976
Changes in Sections	ΔP_E	0.0210
Risers: 8 - 6" STD Wall Pipes		
Entry and Exit	ΔP_S	1.264
Friction in Pipes	ΔP_F	0.8587
Bends	ΔP_B	1.914
Changes in Sections	ΔP_E	~~0.0773~~
Shell	ΔP_{Sh}	0.15
Steam Drum Assumed		0.75
Total Pressure Losses		6.014
Available Head From Table 2.1		6.042 psi

Summary of pressure losses in WHE circuits with recirculation rate of 25.

CHAPTER 3

CHAPTER 3

ASSESSMENT OF FLOW INDUCED TUBE VIBRATION

In the operation of tubular heat exchangers, vibration of the tubes can be induced by fluid flowing over the tube array either in cross flow or in axial flow.

Tubes in cross flow have worn through and failed due to oscillatory contacts with adjacent tubes [5,6] or with support bars [7]. Fatigue failures of tubes at clamping locations have also been reported [8]. Tubes experiencing axial flow are also subject to flow induced vibration [9,10]. While some of the excitation can be attributed strictly to the fluid flowing parallel to the tubes, Paidousis [10] suggests that the cross flow components that also exist in any real axial flow situation can have significant effect on vibration amplitude. Therefore cross flow excitation mechanisms are usually the dominant cause of tube vibration.

The objective of a manufacturer of heat exchange equipment is to provide assurance that a given design will perform reliably. The approach presented below is a method which will permit us to accomplish this objective. There certainly are other valid approaches which could be considered as alternatives.

3.1 APPROACH TO ANALYSIS

Failure of a heat exchanger from tube vibration can result from three basic mechanisms. The first is mechanical wear of the tube caused by rubbing at the support or impacting with adjacent tubes. The thinning of the tube wall can result in rupture from operating pressure. The second mechanism, fretting corrosion (or fretting) is a result of mechanical wear and motion which is so limited that the corrosion debris is not removed from the area of contact and metallurgically accelerates the attack on the tube wall. The third mechanism, fatigue cracking, can occur if bending stresses due to vibration are high.

The two causes of these mechanisms are characterized as follows:

1. Relative motion.
2. Environmental effects.

Relative motion is not a concern if (a) the tubes do not impact, (b) the support does not abrade the tube, and (c) high bending stresses do not occur. What emerges here is that all these "if's" relate to design and fabrication factors.

Concurrently, environmental effects are not a problem if, (a) the effects of coolant purity, temperature, flow velocity, and physical characteristics are known, and (b) the effects of the specified material such as surface finish, hardness, surface composition and other material characteristics are known. These again are large "if's" and require experiments and proof tests for resolution.

In the design of this WHE, the use of too many baffles or support plates impedes the flow of water due to natural circulation. Yet some support plates were necessary to facilitate fabrication of the unit, the initial dimensions of which were arbitarily selected. Following are good design guidelines:

(a) If possible, eliminate high velocities (using baffles, distributors, etc.).

(b) If possible, direct the flow parallel to tubes at least in the vicinity of the high velocity head.

(c) Avoid long limber runs of tubes.

(d) Raise the natural frequency of tubes as much as possible (by employing the bars, clips, etc. addition to tube supports).

(e) Tube supports that clamp onto the tubes with minimal clearances are desirable from vibration viewpoint but may not always be acceptable for flow considerations.

We will now quantitatively assess flow induced vibrations.

3.2 CALCULATION OF NATURAL FREQUENCIES OF TUBES

The natural frequencies of the tubes will be calculated using the procedure outlined in the TEMA STANDARDS [11].

The tubes are assumed to be pinned or hinged at the support plate, and clamped at extreme ends at the tube plates as shown in Figs. 3.1 and 3.2.

The tube natural frequency, f_n, in Hertz, is given by

$$f_N = \frac{3.36\ C}{\ell^2} \sqrt{\frac{E\ I}{W}}$$

where C = mode constant as shown in Figs 3.1 and 3.2

ℓ = span length, inches

E = modulus of elasticity

= 27.6×10^6 psi

I = moment of inertia = $0.9628\ in^4$

W = $W_t + W_{fi} + MW_{fo}$ lb/ft

W_t = weight of empty tube

= 5.56 lb/ft

W_{fi} = weight of fluid inside tube

$= 0.00545\ \rho_i\ d_i^{\,2}$

$= 0.00545 \times 0.08598 \times 2.094^2$

$= 0.0021$ lb/ft

W_{fo} = weight of fluid displaced by tube

$= 0.00545\ \rho_o\ d_o^{\,2}$

for conservative assessment

use $\rho_o = 54.95$ lb/ft^3

= density of water

$W_{fo} = 0.00545 \times 54.95 \times 2.5^2$

$= 1.872$ lb/ft

M = added mass coefficient = 2.26

W = 5.56 + 0.0021 + 2.26 x 1.872

= 9.793 lb/ft

d_o = diameter of tube, inches

First mode natural frequency for the two span tubes as shown in Fig.3.1.

$$f_{N_1} = \frac{3.36 \times 49.59}{223.5^2} \sqrt{\frac{27\ 600\ 000 \times 0.9628}{9.793}}$$

$$= 5.5 \text{ Hz}$$

Second mode natural frequency for the two span tubes

$$f_{N_2} = \frac{3.36 \times 72.36}{223.5^2} \sqrt{\frac{27\ 6000\ 000 \times 0.9628}{9.793}}$$

$$= 8.02 \text{ Hz}$$

Now consider the tubes with two support plates, as shown in Fig. 3.2. The solution is a conservative approximation which will provide lower estimate of the actual natural frequencies.

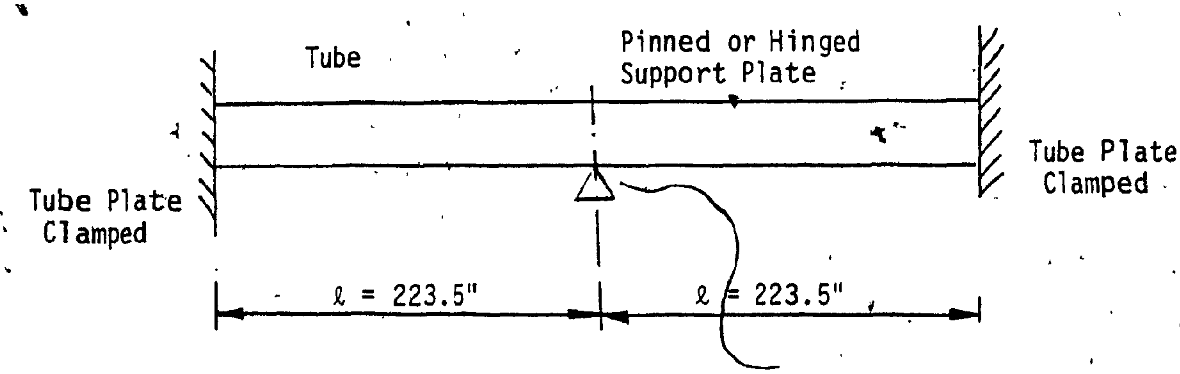


Fig. 3.1: Two Span Tubes
C = 49.59 For First Mode
C = 72.36 For Second Mode

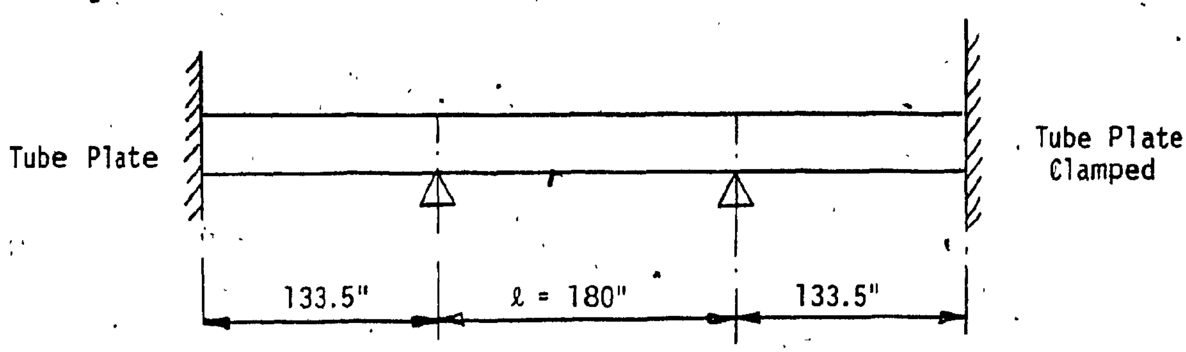


Fig. 3.2: Three Span Tubes
C = 40.52 For First Mode
C = 59.56 For Second Mode

$$f_{N_1} \simeq \frac{3.36 \times 40.52}{180^2} \sqrt{\frac{27\ 600\ 000 \times 0.9628}{9.793}}$$

$$= 6.92 \text{ Hz}$$

$$f_{N_2} \simeq \frac{3.36 \times 59.26}{180^2} \sqrt{\frac{27\ 600\ 000 \times 0.9628}{9.793}}$$

$$= 10.18 \text{ Hz}$$

3.3 ESTIMATE OF FLOW VELOCITIES

For the purpose of vibration analysis we will use a recirculation rate of 30 to obtain a conservative estimate of the flow velocities. Then at exchanger inlet we have liquid flow at point A, Fig. 2.3.

Specific volume at A $= V_A = 0.0182 \frac{ft^3}{lb}$

Density at A $= \rho_A = 54.95$ ft³/lb

Flow velocity, U, parallel to tubes is

$$U = \frac{R\dot{M}_s}{a_f} \quad \frac{V_1}{3600}$$

$\dot{M}_s$ = 50 678 lb/hr of steam from Para. 2.4.2

R = 30

a_f = free flow area parallel to tubes

= 39.06 ft² from Para. 2.4.2.

$$U_{PA} = \frac{30 \times 50\ 678}{39.06} \times \frac{0.0182}{3600} = 0.197 \frac{ft}{sec}$$

Flow velocity normal to tubes U_{NA} is to be based on the minimum shell side or bundle cross flow area, a_s which is given by

$$a_s = \frac{ID \times C'B}{P_T \times 144}$$

where $ID = 100.5$ in

$C' = P_T - d = 3.5 - 2.5 = 1.0$ in

$B = 90$ in

$P_T = 3.5$ in

$$a_s = \frac{100.5 \times 1 \times 90}{3.5 \times 144} = 17.95 \text{ ft}^2$$

To obtain conservative estimate velocity assume all flow is cross flow then

$$U_{NA} = \frac{30 \times 50\ 678 \times 0.0182}{17.95 \times 3600} = 0.43 \frac{\text{ft}}{\text{sec}}$$

Consider now the two phase flow. The specific volume of steam and water mixture leaving WHE is V_{out}.

$$V_{out} = \frac{29}{30} \times 0.0182 + \frac{1}{30} \times 2.616 = 0.1048 \frac{\text{ft}^3}{\text{lb}}$$

$$\rho_{out} = 9.543 \text{ lb/ft}^3$$

$$V_{average} = \frac{1}{2} (V_{out} + V_{in})$$

$$= \frac{1}{2} (0.0182 + 0.1048)$$

$$= 0.0615 \text{ ft/lb}$$

$$\rho_{average} = 16.26 \text{ lb/ft}$$

Average flow velocity parallel to tubes - two phase

$$U_P = 30 \times \frac{50678}{39.06} \times \frac{0.0615}{3600} = 0.665 \text{ ft/sec}$$

Average cross flow velocity - two phase maximum

$$U_N = 30 \times \frac{50678}{17.95} \times \frac{0.0615}{3600} = 1.446 \text{ ft/sec}$$

Two phase velocity near waste heat exchanger outlet

$$U_p = 30 \times \frac{50678}{39.06} \times \frac{0.1048}{3600} = 1.133 \text{ ft/sec}$$

The cross flow velocity near the WHE outlet will be adjusted to allow for the steam-water mixture exiting around the WHE periphery. The flow area will be corrected to allow for the actual configuration.

$$\text{Say flow area} = 2\ a_s = 2 \times 17.95$$

$$= 35.90 \text{ ft}^2$$

$$\therefore U_N = 30 \times \frac{50\,678}{35.90} \times \frac{0.1048}{3600}$$

$$= 1.23 \text{ ft/sec}$$

3.4 VIBRATION EXCITATION MECHANISMS

Generally in cross flow, we consider three basic flow-induced vibration excitation mechanisms, namely:

1. Periodic wake shedding.
2. Fluidelastic instability.
3. Random excitation due to flow turbulence.

The last two mechanisms have been observed in both liquid and two-phase cross flow. Periodic wake shedding resonance is possible in liquid flow but has not been observed in two-phase flow. Either it does not exist or it is dominated by the response to random turbulence.

We will now investigate these flow induced vibration excitation mechanisms.

3.4.1 Periodic Wake Shedding

Periodic wake shedding would generate periodic forces in tube bundles. The periodic formation of vortices downstream of an isolated cylinder is a well understood phenomena called Kármán Vortex Shedding. What happens in closely packed bundles of tubes is not so well understood. Vortex shedding is possible but should be much affected by the close proximity of adjacent and particularly downstream tubes. Buffeting is also possible as a tube may be subjected to periodic forces due to the wake shed by an upstream tube. Whatever the mechanism, if the wake shedding frequency coincides with the ith natural frequency of the tube, resonance may occur in the ith mode. This may be a problem if the vibration response is large enough to control the mechanism of periodic wake shedding. Then the periodic forces become spatially correlated to the mode shape. This phenomenon will be considered in the vibration analysis:

The frequency, f, of the forcing function is given by

$$f = \frac{S_t K_v V}{d_o}$$

where S_t = Strouhal Number

$= 0.35$ for $R_e = 95\,000$

$$\left.\begin{matrix} X_t = 1.4 \\ X_L = 1.4 \end{matrix}\right\} \text{conservative}$$

$K_v V$ = maximum cross flow velocity

$= 1.446$ ft/sec

d_o = tube diameter $= \frac{2.5}{12} = 0.2083$ ft

$$f = \frac{0.35 \times 1.446}{0.2083} = 2.43 \text{ Hz}$$

$$\therefore \frac{f}{f_N} = \frac{2.43}{5.5} = 0.442$$

This is considered to be acceptable.

Magnitude of the forcing function

$$F_S = C_L \rho \frac{V^2}{2g_c} A_T$$

$$C_L = \text{lift coefficient} = 1$$

$$F_S = 1 \times \frac{16.25 \times 1.446^2}{2 \times 32.2} \times \frac{2.5}{12}$$

$$= 0.11 \text{ lb/ft}$$

Maximum dynamic deflection at resonance is

$$X = \frac{K F_S}{M_o \omega_n^2}$$

$$\zeta = \text{damping factor}$$

$$= 0.081 \quad \text{Pettigrew et al, [20]}$$

Magnification factor $K = \frac{X}{X_o}$

$$K = \frac{1}{\sqrt{\left(1 - \frac{\omega}{\omega_n}\right) + \left[2\zeta\left(\frac{\omega}{\omega_n}\right)\right]^2}}$$

at resonance $\omega = \omega_n$

$$\therefore K = \frac{1}{2\zeta} = 6.17$$

$$\therefore X = \frac{6.17 \times 0.11}{9.793\ (2\pi \times 5.5)^2} = \frac{\text{lbf}}{\text{ft}}\ \frac{\text{ft}}{\text{lb}}\ \text{sec}^2 \left[\frac{\text{lb}\ 32.2\ \text{ft}}{1\ \text{lbf}\ \text{S}^2}\right]$$

$$= 0.00187 \text{ ft} = 0.0224 \text{ in}$$

Hence the deflection is negligible. The tube stress corresponding to this deflection is only 220 psi which is also negligible. The fatigue endurance limit of the tube material is about 11 700 psi.

3.4.2 Fluidelastic Excitation

Fluidelastic instabilities are possible in a tube bundle subjected to cross flow when the interaction between the motions of the individual tubes is such that it results in fluid force components that are both proportional to tube displacements and in-phase with tube velocities. Instability occurs when during one vibration cycle the energy absorbed from the fluid forces exceeds the energy dissipated by damping.

For tube bundles subjected to uniform flow over their entire length the following equation applies:

$$\frac{U}{f_N D} = K_c \sqrt{\frac{M_o \delta}{\rho D^2}}$$

where U = reference gap velocities as calculated above ~~in~~ 3.3

f_N = natural frequency

D = tube outside diameter

= 0.2083 ft

K_c = Connors Number which must be less than 3.3 for safe design

δ = Logarthmic Decrement

M_o = mass per unit length = $\frac{9.763}{12}$ = 0.82 lb/in

ρ = fluid density

d_o = tube diameter

$\delta = \frac{C_d}{2M_o f_i}$

C_d = viscous damping coefficient

= $d_c\ C_n$

C_n = normalized viscous damping coefficient

= $400\ \frac{kg}{SM^2}$ for water = $1400\ \frac{kg}{SM^2}$ for two-phase mixtures

Consider first the liquid flow

$$C_d = d_o C_n = 2.5 \times \frac{25.4}{1000} \times 400$$

$$= 25.4 \ \frac{kg}{SM} = 1.41 \ \frac{lb}{in \ sec}$$

$$\delta = \frac{C_d}{2M_o f_N}$$

$$= \frac{1.41}{2 \times 0.82} \times \frac{1}{5.5} \quad \frac{lb}{in \ sec} \quad \frac{in \ sec}{lb}$$

$$= 0.157$$

$$\therefore \text{Critical } U = f_N \ D \ K_c \sqrt{\frac{M_o \delta}{\rho D^2}}$$

$$= 5.5 \times \frac{2.5}{12} \times 3.3 \sqrt{\frac{0.82 \times 0.157 \times 1728}{54.95 \times 2.5^2}}$$

$$= 5.5 \times \frac{2.5}{12} \times 2.66$$

$$= 3.04 \text{ ft/sec}$$

$$\text{Ratio } \frac{U_{CRIT}}{U_{ACT}} = \frac{3.04}{0.43} = 7.08 > 1 \quad \text{OK}$$

Now consider two-phase flow

$$C_d = d_o \ C_n = \frac{2.5 \times 25.4}{1000} \times 1400$$

$$= 88.9 \ \frac{kg}{SM} \left[\frac{2.2 \ lb}{1 \ kg}\right] \left[\frac{1 \ M}{3.281 \ P_T}\right] \left[\frac{1 \ ft}{12 \ in}\right]$$

$$= 4.96 \ \frac{lb}{in \ sec}$$

$$\delta = \frac{C_d}{2M_o f_N} = \frac{4.96}{2 \times 0.82 \times 5.5} = 0.55$$

$$\text{Critical } U = 5.5 \times \frac{2.5}{12} \times 3.3 \sqrt{\frac{0.82 \times 0.55 \times 1728}{16.26 \times 2.5^2}}$$

$$U_c = 10.47 \text{ ft/sec}$$

$$\text{Ratio } \frac{U_c}{U_A} = \frac{10.47}{1.446} = 7.24 > 1 \quad \text{OK}$$

3.4.3 Random Turbulence

Turbulent buffeting is another mechanism sometimes considered in tube vibration analysis. P.R. Owens [21] developed a criterion for predicting damaging vibration by this mechanism using the following equation:

$$\frac{fL}{U} \cdot \frac{T}{d} = 3.05 \left(1 - \frac{d}{T}\right)^2 + 0.28$$

where f = frequency at which most energetic eddies encounter tubes, Hz

L = longitudinal spacing between tubes, ft

U = velocity between tubes, ft/sec

T = transverse spacing between tubes, ft

d = tube diameter, ft

$$\therefore f = \frac{U}{L} \left(\frac{d}{T}\right) 3.05 \left(1 - \frac{d}{T}\right)^2 + 0.28$$

$$= \frac{1.446}{3.5} \times 12 \times \frac{2.5}{12} \times \frac{12}{3.5} \quad 3.05 \left(1 - \frac{2.5}{3.5}\right)^2 + 0.28$$

$$= 1.87 \text{ Hz}$$

$$\frac{f_{TURBULENCE}}{f_{NAT}} = \frac{1.87}{5.5} = 0.34 \qquad \text{OK}$$

3.4.4 Use of Further Published Literature

Some purchasers and designers prefer to use the simplified criteria presented by J.T. Thorngren [22].

This paper gives two equations to assess the so called baffle type damage and the tube collision type damage.

The criteria for baffle type damage is given by the following dimensionless number, N_{BD}.

$$N_{BD} = \frac{D \rho U^2 \ell^2}{F_B S_M g_c A_M B_t}$$

A safe value of N_{BD} would be less than one,

where D = tube diameter = 2.5 in

ρ = fluid density = 16.26 lb/ft^3

U = fluid velocity = 1.446 ft/sec

ℓ = length of tube between supports = 223.5 in

F_B = tube to support clearance factor

= 1

S_M = maximum allowable fatigue stress

= 11,700 psi

g_c = gravitational constant = 32.2 $\frac{ft}{s^2}$

A_M = tube cross sectional metal area

= 1.45 in^2

B_t = support plate thickness = 0.75 in

$$N_{BD} = \frac{2.5 \times 16.26 \times 1.446^2 \times 223.5^2}{1 \times 11{,}700 \times 32.2 \times 1.45 \times 0.75}$$

$$\text{in} \ \frac{lb}{ft^3} \ \frac{ft^2}{s^2} \ \frac{in}{lb_F} \ \frac{in^2}{in^2} \ \frac{sec^2}{ft} \ \frac{1}{in} \left[\frac{ft^2}{144\ in^2}\right]$$

$= 0.072 < 1.0$ OK

The criteria for collision type damage is given by

$$N_{CD} = \frac{0.625\ D\ \rho\ U^2\ \ell^4}{F_B{}^4\ g_c\ A_M\ (D^2 + d_i{}^2)\ C_T\ E}$$

where symbols are as above and

d_i = tube inside diameter = 2.1 in

C_T = minimum clearance between tubes = 1 in

E = 27.6 x 10^6 psi

Again, safe design criterion is for N_{CD} to be less than one.

$$N_{CD} = \frac{0.625 \times 2.5 \times 16.26 \times 1.446^2 \times 223.5^4}{1 \times 32.2 \times 1.45\ (2.5^2 + 2.1^2)\ 1 \times 27\ 600\ 000}$$

$$\text{in}\ \frac{lb_F}{ft^3}\ \ \frac{ft^2}{s^2}\ \ \frac{in^4}{in^2}\ \ \frac{s^2}{ft}\ \ \frac{1}{in^2}\ \ \frac{1}{in}\ \ \frac{in^2}{lb_F}\ \ \left[\frac{ft^2}{144\ in^2}\right]$$

$= 0.067 < 1$ OK

3.5 CONCLUSIONS FROM VIBRATION ANALYSIS

The calculations presented in Chapter 3 gives assurance that no flow induced vibration problem is expected when the WHE unit is operated in accordance with the specified normal operating conditions.

CHAPTER 4

CHAPTER 4

CODE REQUIREMENTS AND BASIC DESIGN CALCULATIONS

Heat exchangers and other pressure vessels were originally constructed with rivetted joints. During the past fourty years, fusion welding has been developed to the degree that it is now completely acceptable to the Provincial Department of Labour as a method of joining pressure parts. The continuing improvement of welding techniques has resulted in fusion welding being adopted by industry as virtually the only method of joining parts for pressure vessels. In most larger and more modern plants, fabricating pressure vessels, the main weld seams are now made with automatic welding machines. This type of welding equipment greatly reduces fabrication time and costs and also improves the quality of the welds.

Every Province of Canada has legally adopted Canadian Standards Association (C.S.A.) Standard No. B51 which embodies the American Society of Mechanical Engineers (A.S.M.E.) Boiler and Pressure Vessel Code, Sections I to XI inclusive, to govern the design and construction of power boilers and pressure vessels. Section VIII, Division I of the ASME Code entitled "Pressure Vessels" is the section governing the design and construction of this waste heat exchanger. Standards of Tubular Exchanger Manufacturers Association (TEMA) Class R was also implemented, while the piping design and fabrication was governed by the Refinery Piping Code ANSI B 31-3 [14].

Based on the requirements of Section VIII of the ASME Code and current construction practices, the principal components of the waste heat exchanger, as shown on drawings enclosed in Appendix C, may be discussed as follows.

4.1 MATERIALS

Part UG of the code gives the general requirements for materials and allowable stresses are given in Table UCS 23 in Subsection C [13]. The materials selected are summarized on drawing GN 1.

4.2 WHE PRESSURE PART DESIGN

The design data is given on drawing GN 1.

4.2.1 Shell: Subjected to Internal Pressure

The shell is usually formed by rolling flat plates to the required diameter and welding the seams. The formula specified in Paragraph UG 27 C of the code to determine the minimum thickness of the shell plate is

$$t = \frac{PR}{SE - 0.6\,P} + C'$$

where t = minimum thickness of the shell in inches

P = design internal pressure = 200 psig

S = allowable stress for material

= 17 500 psi for SA 516 GR 70

R = inside radius of the shell in inches = 50.50 in

E = joint efficiency

= 1.0 for full penetration double butt joint, fully radiographed

C' = corrosion allowance = 0.125 in as specified by the purchaser.

$t = 0.5812 + 0.125 = 0.7063$ in

$\frac{3}{4}$ in thick plate was used for the exchanger shell.

This simple formula is derived from the Lamé equation. This formula was used for determining the thickness required for all cylindrical sections, such as shell, nozzles, tubes, subjected to interal pressure.

4.2.2 Tubes Subjected to External Pressure

The hot gases passing through the tubes are at 15.3 psig while steam and water on the outside of tubes are at a design pressure of 200 psig. The equations of membrane theory are not valid when the pressure is applied on convex side of a shell of revolution. Instead of the tensile hoop stress, the ability of the shell to withstand local buckling becomes the governing factor to prevent collapse. The rules for determining thickness of shells and tubes under external pressure are given in Paragraph UG 28 of the code.

The tubes are fastened to the tube plates by roller expanding and seal welding. The actual tubes used are 470 - 2½" o/D x 0.200" minimum thickness.

Then using

$$\frac{D_o}{t} = \frac{2.5}{0.2} = 12.5 > 10 .$$

$$\frac{L_T}{D_o} = \frac{450}{2.5} = 180$$

A = 0.0075 from Fig. UGO 28.0 in Appendix V of code.

B = 13 000 psi for SA 516 GR70 again from Appendix V.

Allowable external pressure $P_a = \dfrac{4B}{3\dfrac{D_o}{t}}$

$$= \frac{4 \times 13\ 000}{3 \times 12.5} = 1386 \text{ psi} > 200 \text{ psi}$$

Hence, $2\frac{1}{2}$ $^{O}/_{D}$ x 0.200" thk tubes are acceptable for the design external pressure.

4.2.3 Openings and Reinforcement

The code gives rules for providing openings in pressure vessels in Paragraphs UG-36 through UG-46. The general objective in providing openings in pressure vessels is, of course, to make the opening in such a way that the strength of the vessel is not reduced. For very small, widely scattered openings, nothing needs to be done to prevent a significant reduction in strength and UG-36 permits such openings up to certain sizes in certain thicknesses of vessels. For larger openings, however, material must be added to the vessel around the opening to prevent a reduction in static strength.

The general code method for deciding how much material should be added is the so-called 100% reinforcement method. This means that the amount of material to be added, as viewed in a cross section taken through the hole, must be at least equal to the amount of material removed from the vessel wall in providing the hole. Obviously, to be of value in strengthening the hole, this material must be located near the hole, and not on the other side of the vessel. Thus the code also gives rules for how close to the hole the reinforcement material must be located both in directions parallel and perpendicular to the pressure vessel wall.

Figure 4.1 is a copy of Fig. UA 280 from the code which summarizes the foregoing reinforcement design criteria.

As an example, consider now the 24 in manway on vessel shell:

Required neck thickness $t_{rn} = \frac{PR}{SE - 0.6\,P} + C'$

$$t_{rn} = \frac{200 \times 11.6875}{17\,500 - 0.6 \times 200} + 0.125 = 0.2595 \text{ in}$$

actual plate used $t_n = 0.5$ in

Required area of reinforcement = $A = d \times t_r \times F$

$d = 23.375$ in , $F = 1$

$t_r = 0.5783$ from section 4.2.1

$A = 23.375 \times 0.5783 \times 1$

$= 13.52 \text{ in}^2$

Available area for reinforcement:

$A_1 = (t - t_r)\, d = (0.625 - 0.5783)\, 23.125$

$= 1.07 \text{ in}^2$

$A_2 = (t_n - t_{rn})\, 5t = (0.5 - 0.2595)\, 5 \times 0.5$

$= 0.601 \text{ in}^2$

$A_3 = 0$

$A_4 = 2 \times \frac{1}{2} \times \frac{3}{8} \times \frac{3}{8} = 0.1406 \text{ in}^2$

$A_5 = (D_p - d - 2t_n)\, t_e$

$= (45.125 - 23.375 - 2 \times 0.5) \times 0.75$

$= 15.56 \text{ in}^2$

$A_1 + A_2 + A_3 + A_4 + A_5 = 17.37 \text{ in}^2 > A = 13.52 \text{ in}^2$

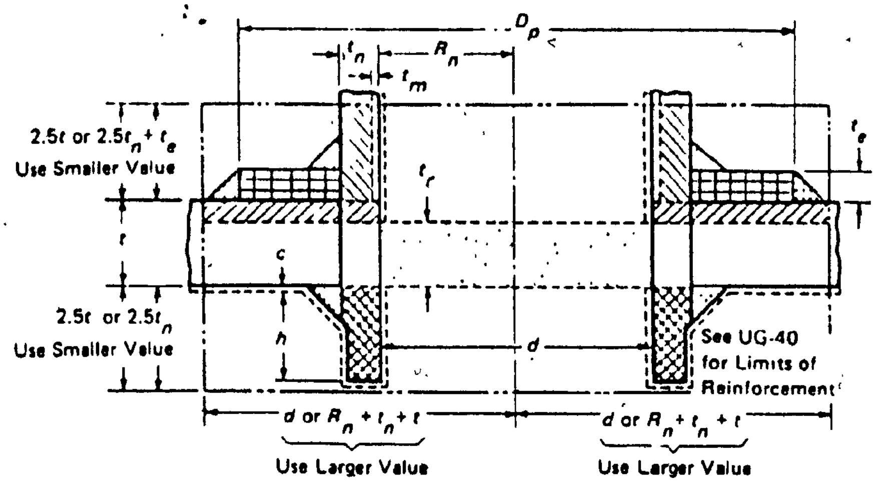
Similarly the reinforcement required for the downcomer and riser nozzles was provided all in the shell thickness as A_1.

$A_1 = A$

$(t - t_r)\, d = d\, t_r$

$t = 2t_r$

$t = 2 \times 0.5783 = 1.1566$



WITHOUT REINFORCING ELEMENT

$= A = d \times t_r \times F$ = area of reinforcement required

$= A_1 \begin{cases} = (E_1 t - F t_r)(d - R_n)\,2 = (E_1 t - F t_r)\,d \\ \text{or} \\ = (E_1 t - F t_r)(R_n + t_n + t - R_n)\,2 = (E_1 t - F t_r)(t_n + t)\,2 \end{cases}$ Larger value is area of shell available for reinforcement.

$= A_2 \begin{cases} = (t_n - t_{rn})\,2.5\,t \times 2 = (t_n - t_{rn})\,5\,t \\ \text{or} \\ = (t_n - t_{rn})\,2.5\,t_n \times 2 = (t_n - t_{rn})\,5\,t_n \end{cases}$ Smaller value is area of nozzle wall available for reinforcement.

$= A_3 = (t_n - c)\,h \times 2 = (t_n - c)\,2h$*

$= A_4$ = Area of welds

$= A_5$ = Area added by reinforcing element

If $A_1 + A_2 + A_3 + A_4 \geq A$	Opening is adequately reinforced.
If $A_1 + A_2 + A_3 + A_4 < A$	The difference must be supplied by reinforcing element or otherwise.

WITH REINFORCING ELEMENT

A, A_1, A_3, A_4, same as without reinforcing element.
With a reinforcing element, 2.5 t_n is measured from the top surface of the reinforcing element.
A_2 becomes the smaller of $(t_n - t_{rn})\,5t$ or $(t_n - t_{rn})(2.5\,t_n + t_e)\,2$.
Area of reinforcing element $= (D_p - d - 2\,t_n)\,t_e = A_5$.
If $A_1 + A_2 + A_3 + A_4 + A_5 \geq A$ opening is adequately reinforced.

*The nozzle projection will not corrode back of any attaching fillet, hence the term $(t_n - c)$ is slightly conservative.

EXAMPLE OF A REINFORCED OPENING
(This Figure Illustrates a Common Nozzle Configuration and Is Not Intended to Prohibit Other Configurations Permitted by the Code.)

Fig. 4.1

With corrosion allowance of 0.125 in

Required shell thickness = 1.2816 in

Hence the use of $1^{5}/_{16}$ in thk subshells, which are upper and lower portions of the WHE shell where downcomer and riser nozzles are attached.

4.2.4 Flange Design

Flanges and flanged connections are very important pressure vessel components. Flanges permit the easy and quick assembly and disassembly of the heat exchanger sections for cleaning, inspection, etc and as flanged nozzles, permit the connections of piping, instruments and mechanical parts to the WHE vessel.

For nozzles up to 24 inch nominal size, standard flanges with dimensions and pressure ratings per ANSI B 16.5 are normally used. They permit the attachment of piping fabricated by others without the provision of special mating flanges. The ANSI standard lists several pressure classes from 150 psig to 2500 psig.

For in-between pressure ratings, as in this instance for the waste heat exchanger, it is usually more economical to select the next higher pressure class rather than to design a flange for the actual condition.

Shell flanges and nozzles flanges over 24 in nominal size are usually designed for the actual conditions required. The ASME code provides a design procedure based on the "Beam on an Elastic Foundation" method combined with the deflection of an annular plate. This method is based on the work of Waters, Wesstrom, Rossheim and Williams and is identical with the method used to establish the dimensions of standard ANSI flanges. A booklet by Taylor Forge Company: "Modern Flange Design"

[16] contains the ASME method and is very useful as it provides a data sheet for organized computation of several flange configurations.

We will now present calculations for the design of the blind cover for a 24 in manway and the main flanges on the waste heat exchanger shell.

A. <u>24 in Manway Cover</u>:

The cover will be designed to match the mating flange which is 24 in - 300 # slip on raised face flange to ANSI B 16.5 and SA 105 material.

Cover Material	: SA 516 GR 70
Bolting Material	: SA 193 GR B 7
Gasket	: Asbestos Jacketted Stainless Steel
Design Pressure	$P = 200$ psig
Design Temperature	$T = 650°F$
Allowable Stress	$S = 17\,500$ psi
Bolt Allowable Stress	$S = 25\,000$ psi
Gasket Seating Stress	$Y = 9000$ psi
Gasket Factor	$M = 3.75$
Gasket Width	$N = 1.125$ in
Gasket Mean Diameter	$G = 25.875$ in

$b_0 = 0.5N = 0.5625$ $\quad b = \sqrt{b_0} = 0.375 \quad \because b_0 > 0.25$

For details see Fig. 4.2.

1. Required bolt load for operating condition

$$W_{M_1} = \frac{\pi}{4} G^2 P + 2b\,\pi\,G\,PM$$

$$= \frac{\pi}{4} \times 25.875^2 \times 200 + 2 \times 0.375\pi \times 25.875 \times 200 \times 3.75$$

$$= 105\,168 + 45\,725$$

$$= 150\,895 \text{ lbf}$$

Bolt area required = $\frac{150\ 893}{25\ 000}$ = 6.04 in^2

2. For gasket seating condition

$$W_{M_2} = \pi bGY = \pi \times 0.375 \times 25.875 \times 9000$$
$$= 274\ 350 \text{ lbf}$$

Bolt area required = $\frac{274\ 350}{25\ 000}$ = 10.97 in^2

bolt area available with 24-1½in bolts

$$= 24 \times 1.405 = 33.72 \text{ in}^2$$

Cover thickness for operating condition, t_{oc}

$$t_{oc} = d_c \sqrt{\frac{CP}{SE} + \frac{1.94\ WM_1 \times h_g}{S \times d_c^3}}$$

$$= 25.875 \sqrt{\frac{0.3 \times 200}{17\ 500} + \frac{1.9 \times 150\ 893 \times 3.0625}{17\ 500 \times 25.875^3}}$$

$$= 2.058 \text{ in}$$

$$t_{oc} = 2.058 + 0.125 = 2.183 \text{ in}$$

Cover thickness for gasket seating condition t_{Gc}

$$t_{Gc} = d_c \sqrt{\frac{1.9\ W_{ATM} \times h_g}{S\ d_c^3}}$$

$$W_{ATM} = \frac{10.97 + 33.72}{2} \times 25\ 000 = 558\ 625 \text{ lbf}$$

$$t_{Gc} = 25.875 \sqrt{\frac{1.9 \times 558\ 625 \times 3.0625}{17\ 500 \times 25.875^3}} = 2.68 \text{ in}$$

$$t_{Gc} = 2.68 + 0.125$$

Use t = 2⅞ in   OK

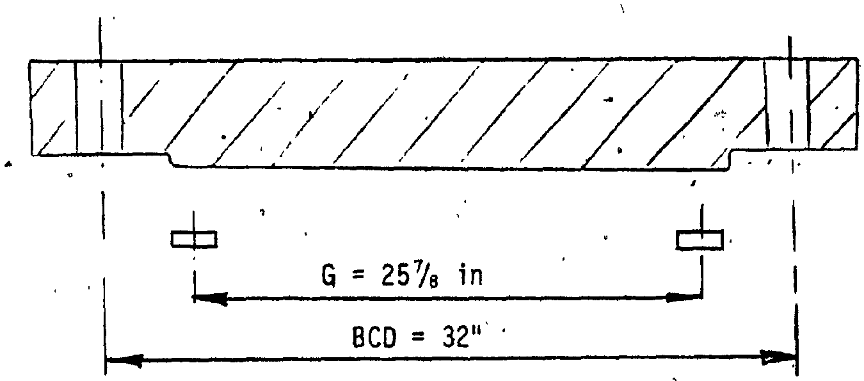


Fig. 4.2: Manway Cover

B. Design of WHE Main Flanges

Case 1: Assessment of main flanges for internal design pressure, P, of 40 psi only. Solution for this case is shown in Table 4.1, which also shows the Code allowable stresses for pressure loading only.

Case 2: In this case, we will present an acceptable method for calculating stresses in flanged joint subjected to internal pressure and external moments and forces.

The design pressure used for the calculation of loads in the flanged joint by equations in Case 1 shall be replaced by a flange design pressure, $P_{FD} = P + P_{EQ}$, where P is the maximum operating pressure and P_{EQ} is an equivalent pressure to account for the moments and forces acting on the flanged joint due to weight and thermal expansion of the piping. The equivalent pressure, P_{EQ}, shall be determined by the equation

$$P_{EQ} = \frac{16\ M_E}{\pi G^3} + \frac{4\ F_Y}{\pi G^2}$$

where

M_E = bending moment applied to the joint due to weight and thermal expansion of the piping, in lbf

F_Y = axial force applied to the joint due to weight and thermal expansion of piping, lbf

G = diameter at location of effective gasket load reactions.

Specified conditions:

Design Pressure = 40 psi

Maximum Operating Pressure = 15.3 psi

Table 4.1: Design of Gas Inlet and Outlet Flanges on WHE for Design Pressure of 40 psig.

WELDING NECK FLANGE DESIGN — SHEET A

DESIGN CONDITIONS		GASKET and BOLTING CALCULATIONS		FROM Fig. UA 49.1 and UA 49.2
Design Pressure, P	40 psi	Gasket Details: 104.125" I/D x 105.125" O/D x 3/16 thk SS410 flat Metal Jckd. Asbestos	Facing Details: 105.75" O/D x 1/8 Raised Face	N = 0.5
Design Temperature	650°F			b = 0.25
Flange Material	SA 516 GR 65			y = 9000
Bolting Material	SA 193 GR B7			m = 3.75

Corrosion Allowance		0.125	$W_{m2} = b\pi G y$ = 739 551	A_m = greater of W_{m2}/S_a or W_{m1}/S_b = 29.58 in²
Allowable Stresses — Flange	Design Temp., S_{fo}	16 300	$H_p = 2b\pi G m P$ = 24 652	A_b = 72 x 0.419 = 30.17 in²
	Atm. Temp., S_{fa}	16 300	$H = G^2\pi P/4$ = 343 891	$W = .5(A_m + A_b)S_a$ = 746 850
Bolting	Design Temp., S_b	25 000	$W_{m1} = H_p + H$ = 368 543	W_{m1} =
	Atm. Temp., S_a	25 000	Gasket Width Check (Raised Face ONLY): $N_{min} = A_b S_a / 2y\pi G$ = 0.127	

CONDITION	LOAD	x	LEVER ARM	=	MOMENT
Operating	$H_D = \pi B^2 P/4$ = 318 889		$h_D = R + .5g_1$ = 2.4688		$M_D = H_D h_D$ = 787 273
	$H_G = W_{m1} - H_p$ = 24 652		$h_G = .5(C - G)$ = 1.125		$M_G = H_G h_G$ = 27 734
	$H_T = H - H_D$ = 25 002		$h_T = .5(R + g_1 + h_G)$ = 2.09375		$M_T = H_T h_T$ = 52 348
					M_o = 867 355
Gasket Seating	$H_G = W$ = 746 850		$h_G = .5(C - G)$ = 1.125		M_g = 840 207

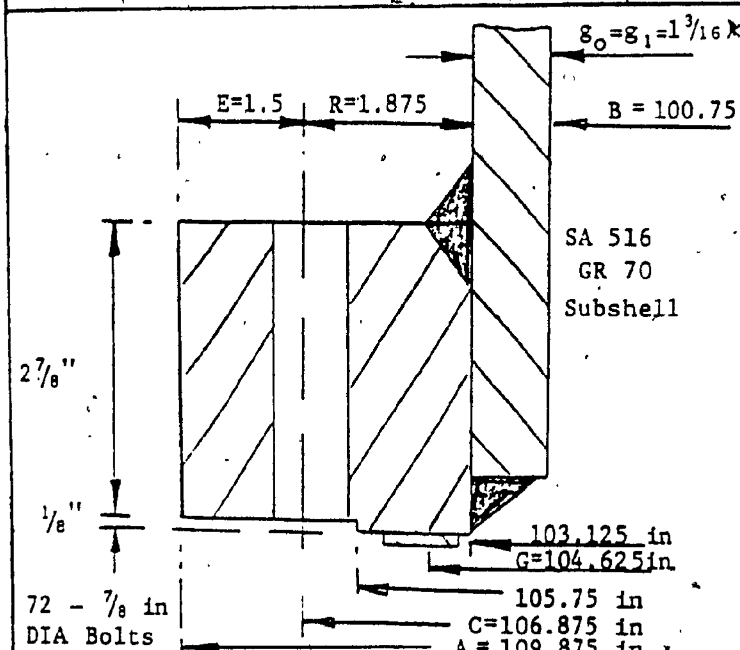
Allowable Stress	STRESS CALCULATION — Operating	Conditions (use M_o)		
1.5 S_{fo}	Long. Hub, $S_H = f M_o / \lambda g_1^2$	8684		
S_{fo}	Radial Flg., $S_R = \beta M_o / \lambda t^2$	1812		
S_{fo}	Tang. Flg., $S_T = (M_o Y / t^2) - Z S_R$	376		
S_{fo}	greater of .5($S_H + S_R$) or .5($S_H + S_T$)	5248		

Allowable Stress	STRESS CALCULATION — Gasket Seating (use M_g)			
1.5 S_{fa}	Long. Hub, $S_H = f M_g / \lambda g_1^2$	8413		
S_{fa}	Radial Flg., $S_R = \beta M_g / \lambda t^2$	1756		
S_{fa}	Tang. Flg., $S_T = (M_g Y / t^2) - Z S_R$	363		
S_{fa}	greater of .5($S_H + S_R$) or .5($S_H + S_T$)	5085		

SHAPE CONSTANTS		From design table 2 and designs charts 1, 2 & 5	
$K = A/B$	1.091	h/h_o	0.549
T	1.88	F	0.908
Z	11.52	V	0.55
Y	22.22	f	1
U	24.41	$e = F/h_o$	0.083
g_1/g_o	1	$d = \frac{U}{V} h_o g_o^2$	684.7
$h_o = \sqrt{B g_o}$	10.94		

OTHER STRESS FORMULA FACTORS			
t (assumed)	3.0		
$\alpha = te + 1$	1.249		
$\beta = 4/3\, te + 1$	1.332		
$\gamma = \alpha/T$	0.664		
$\delta = t^3/d$	0.0394		
$\lambda = \gamma + \delta$	0.703		
$M = M_o/B$	8 609		
$M = M_g/B$	8340		

If bolt spacing exceeds $2a + t$, multiply M_o and M_g in above equations by $\sqrt{\frac{\text{Bolt spacing}}{2a + t}}$.



Computed JSK   Date Feb. 14, 79

Checked   Number

SOURCE	F_y Axial Force lbf	F_z Shear Force lbf	M_x Moment KIP in
Deadweight	-29000	-1930	-381
Thermal	3030	-16400	-7199
Earthquake	2240	-6530	-1764
T + D + EQ	-23 730	-24 860	-9345

Table 4.2: Specified External Loads at Flange Joint.

$$\therefore \; P_{EQ} = \frac{16 \times 9345 \times 10}{\pi \times 104.625^3} \times \frac{4 \times 23\,730}{\pi \times 104.625^2}$$

$$= 41.55 + 2.76$$

$$= 44.3 \text{ psi}$$

$$P_{FD} = P + P_{EQ}$$

$$P_{FD} = 15.3 + 44.3$$

$$= 59.6 \text{ psi}$$

It was decided to use P_{FD} = 60 psi. The calculated stresses using this pressure are shown in Table 4.3.

The longitudinal hub stress, S_H, is revised to include the primary axial membrane stress as follows:

$$S_H = \frac{f\,M_o}{\lambda g_1{}^2 B} + \frac{PB}{4g_o}$$

$$= \frac{1 \times 1\,301\,031}{0.703 \times 1.1875^2 \times 100.75} + \frac{15.3 \times 100.75}{4 \times 1.1875}$$

Design of Gas Inlet Flange on WHE For
Operating Pressure and External Moments
i.e. Equivalent Pressure of 60 psig.

TABLE 4.3

WELDING NECK FLANGE DESIGN — SHEET A

DESIGN CONDITIONS		GASKET and BOLTING CALCULATIONS		FROM Fig UA 49.1 and UA 49.2
Design Pressure, P	60 psig	Gasket Details: 104.125" I/D x 105.125" O/D x 3/16 thk SS 410 Flat Metal Jckd. Asbestos	Facing Details: 105.75" O/D x 1/8" Raised Face	N = 0.5
Design Temperature	650°F			b = 0.25
Flange Material	SA 516 GR 65			y = 9000
Bolting Material	SA 193 GR B7			m = 3.75

Corrosion Allowance		0.125	$W_{m2} = b\pi G y$ = 739 551	A_m = greater of W_{m2}/S_a or W_{m1}/S_b = 29.58
Allowable Stresses — Flange	Design Temp, S_{fo}	16 300	$H_p = 2b\pi G m P$ = 36 978	A_b = 72 x 0.419 = 30.17 in²
	Atm Temp, S_{fa}	16 300	$H = G^2\pi P/4$ = 515 836	$W = .5(A_m + A_b)S_a$ = 746 850
Allowable Stresses — Bolting	Design Temp, S_b	25 000	$W_{m1} = H_p + H$ = 552 814	W_{m1} =
	Atm Temp, S_a	25 000	Gasket Width Check (Raised Face ONLY), $N_{min} = A_b S_a / 2y\pi G$ = 0.127	

CONDITION	LOAD	x	LEVER ARM	=	MOMENT
Operating	$H_D = \pi B^2 P/4$ = 478 334		$h_D = R + .5g_1$ = 2.4688		$M_D = H_D h_D$ = 1 180 911
	$H_G = W_{m1} - H$ = 36 978		$h_G = .5(C - G)$ = 1.125		$M_G = H_G h_G$ = 41 600
	$H_T = H - H_D$ = 37 502		$h_T = .5(R + g_1 + h_G)$ = 2.09375		$M_T = H_T h_T$ = 78 520
					M_o = 1 301 031
Gasket Seating	$H_G = W$ = 746 850		$h_G = .5(C - G)$ = 1.125		M_g = 840 207

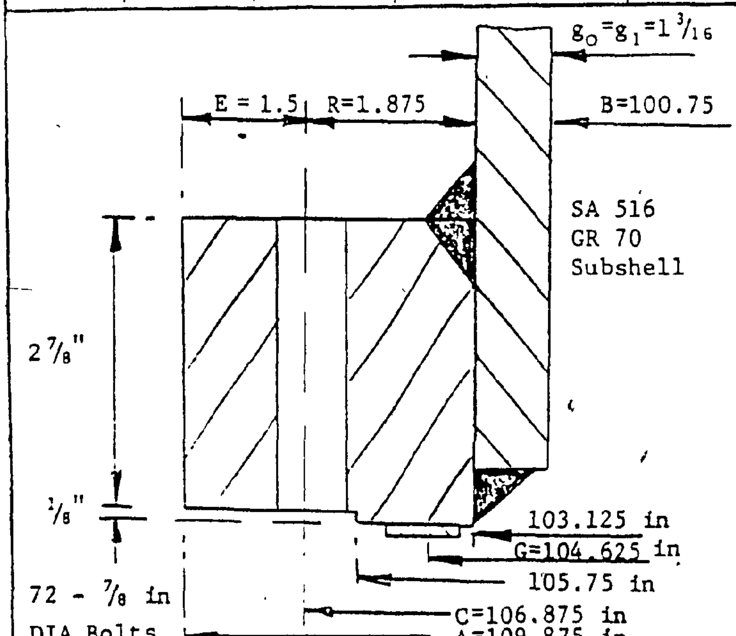
Allowable Stress	STRESS CALCULATION — Operating (use M_o)			
1.5 S_{fo}	Long. Hub, $S_H = f M_o/\lambda g_1^2 B$	13 027		
S_{fo}	Radial Flg, $S_R = \beta M_o/\lambda t^2 B$	2719		
S_{fo}	Tang. Flg, $S_T = (M_o Y/t^2 B) - Z S_R$	538		
S_{fo}	greater of $.5(S_H + S_R)$ or $.5(S_H + S_T)$	7873		

Allowable Stress	STRESS CALCULATION — Gasket Seating (use M)			
1.5 S_{fa}	Long. Hub, $S_H = fM/\lambda g_1^2 B$	8413		
S_{fa}	Radial Flg., $S_R = \beta M/\lambda t^2 B$	1756		
S_{fa}	Tang. Flg., $S_T = (MY/t^2 B) - Z S_R$	364		
S_{fa}	greater of $.5(S_H + S_R)$ or $.5(S_H + S_T)$	5085		

SHAPE CONSTANTS (From design table 2 and design charts 1, 2 & 5)			
K = A/B	= 1.091	h/h_o	= 0.549
T	= 1.88	F	= 0.908
Z	= 11.52	V	= 0.55
Y	= 22.22	f	= 1
U	= 24.41	$e = F/h_o$	= 0.083
g_1/g_o	= 1	$d = \frac{U}{V} h_o g_o^2$	= 684.7
$h_o = \sqrt{B g_o}$	= 10.94		

OTHER STRESS FORMULA FACTORS	
t (assumed)	3.0
$\alpha = te + 1$	1.249
$\beta = 4/3\, te + 1$	1.332
$\gamma = \alpha/T$	0.664
$\delta = t^3/d$	0.0394
$\lambda = \gamma + \delta$	0.703
$M = M_o/B$	12913
$M = M_g/B$	8340

If bolt spacing exceeds 2a + t, multiply M_o and M_g in above equations by $\sqrt{\frac{\text{Bolt spacing}}{2a + t}}$



Computed JSK Date

Checked Number

$= 13\ 027 + 325$

$= 13\ 352$ psi

Radial stress in flange, S_R, = 2719 psi

Tangential stress in flange, S_T, = 538 psi

The allowable stress limits for the above combined loads in Case 2 are

S_H not greater than 1.5 S = 1.5 x 17 500 = 26 250 psi

S_R not greater than 1.5 S = 1.5 x 16 300 = 24 450 psi

S_T not greater than 1.5 S = 1.5 x 16 300 = 24 450 psi

Shear stress in shell $= \frac{F_Z}{2\pi r t} = \frac{24.860}{2\pi \times 100.75 \times 1.1875}$

$= 0.033$ KSI

Shear stress in bolts $= \frac{F_Z}{A_B} = \frac{24.860}{30.17}$

$= 0.823$ KSI

4.3 DESIGN OF WHE SUPPORTS

The outline of the WHE supports are given in Figs. 4.3 and 4.4. The properties of the support rings are given in Table 4.5.

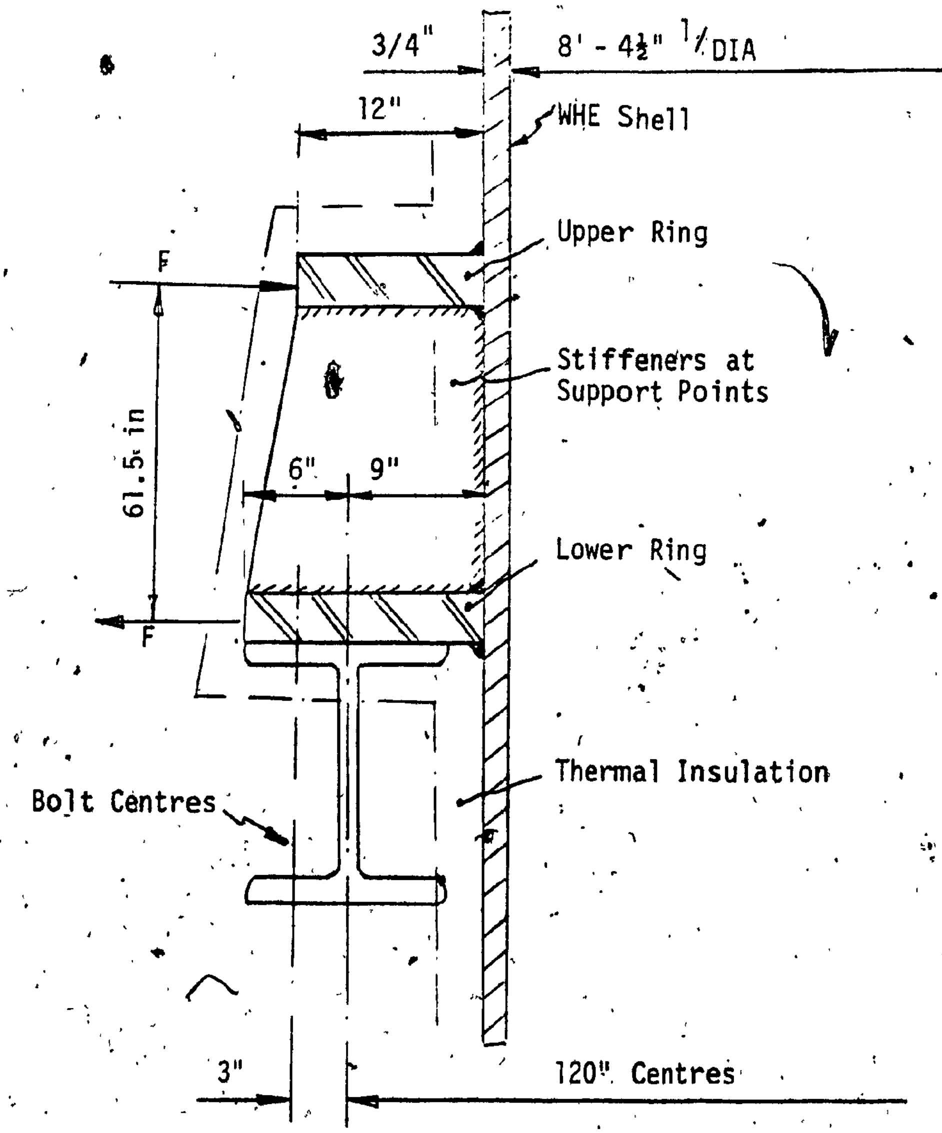
4.3.1 Specified Loads

The support rings provide an eight point support subjected to the following loading:

1. Operating Weight

   Total maximum weight = 333 KIP = 42 KIP per support point

   Lower operating weight = 226 KIP = 28 KIP per support point



Both rings are 1½" thk
All eight stiffeners are each $1\frac{5}{16}$ in thk

Fig. 4.3: Details of WHE Ring Support

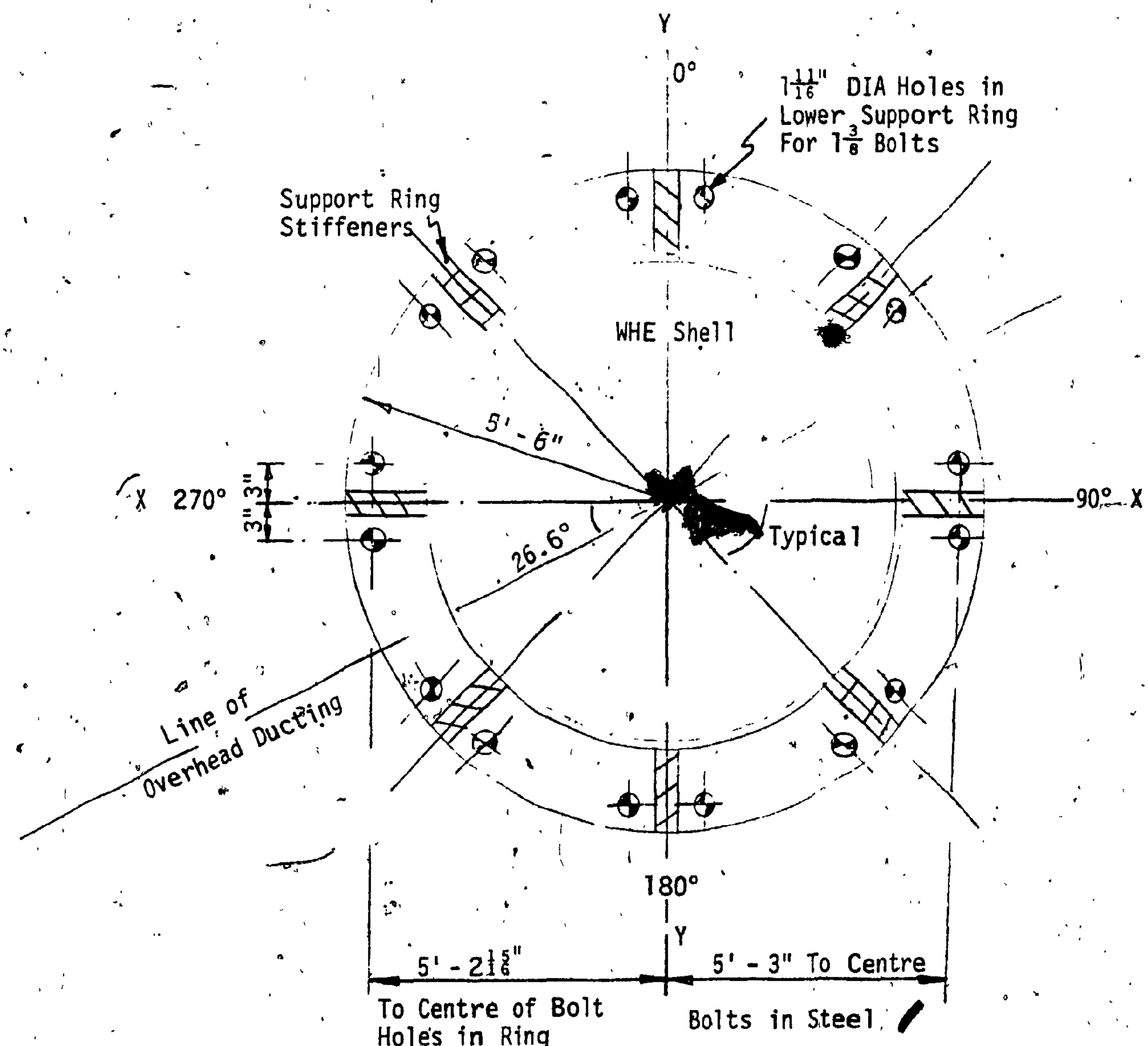


Fig. 4.4: Plan on Lower Support Ring

2. Wind Load

From National Building Code of Canada -

$P_W = q\ c_e\ c_g\ c_p$

or $F_W = q\ c_e\ c_g\ c_n\ A_p$ for cylindrical vessel

where P_W = the design external pressure acting statically and in a direction normal to the surface

F_W = total shear force

q = the reference velocity pressure = 7.8 psf

c_e = the exposure factor = 1.5 conservative

c_g = the gust factor = 2.0

c_p = external pressure coefficient

c_n = cross section or roughness coefficient

A_p = projected area = $d_o \times h$

d_o = diameter = 9.166 ft

h = height = 63.5625 ft

$\therefore\ F_W = 7.8 \times 1.5 \times 2.0 \times 0.9 \times 9.1666 \times 63.5625$

$= 12\ 271$ lbf

Take this as shear only since support bracket is just above the centre of the WHE.

Assuming four brackets are resisting wind shear, then load per bracket is

$$V_W = \frac{F_W}{4} = \frac{12\ 271}{4} = 3068 \text{ lbf}$$

Say V_W = 4 KIP

3. Effects of Earthquakes

Again, using NBC Section 4.1.9, Paragraph 12

$V_{PT} = A_c\ S_p\ W_p$

V_{PT} = lateral force

A_c = assigned horizontal design acceleration

= 0.04

S_p = horizontal force factor = 3

W_p = weight of WHE = 333 KIP maximum

$V_{PT} = 0.04 \times 3 \times 333$

= 39.96 KIP

Again, assuming four brackets effective

$V_p = \frac{39.96}{4} = 9.99$ KIP

Say V_p = 10 KIP This governs when compared with wind loading.

4. Friction Force Resisting Thermal Expansion

$F_N = \mu W_p$

$= 0.35 \times 333 = 116.55$ KIP

= 14.6 KIP per support point.

Say F_N = 15 KIP per bracket

5. The specified external loads from the overhead ducting at the WHE supports are as shown in Table 4.4.

Positive values of F_y indicates an uplift.

SOURCE	F_y Axial Force lbf	F_z Shear Force lbf	M_x Moment KIP in
Weight		-1930	-820
Thermal	3030	-16 400	-10 930
Earthquake	2240	- 6 530	- 3250
T + D + EQ	5270	-24 860	-15 000

Table 4.4: Specified External Loads at WHE Support Level

4.3.2 <u>Configuration of Supports</u>

The details of WHE ring supports are given on drawings E1 and 1.

The elevation of support ring is shown in Fig. 4.3 and plan on the lower support ring is given in Fig. 4.4

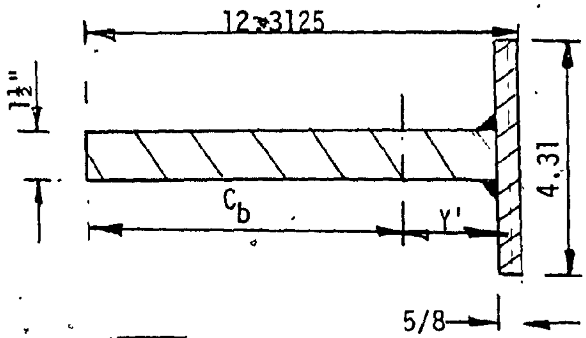
The calculation of properties of upper and lower support rings are shown in Table 4.5

4.3.3 <u>Calculation of Stresses</u>

A. <u>Stress Due to Deadweight and Shear Loads</u>

From Ref. [15] "Design of Welded Structures", by Omer Blodgett, Section 6.6.3, we have forces (f_1) normal to the shell which set up tangential tensile forces (T) and bending moments (M_r) in the ring of the shell, Fig. 4.3, as listed in Table 4.6.

UPPER SUPPORT RING



$$e = \sqrt{\frac{t_s r_c}{2}} = \sqrt{\frac{0.625 \times 50.375}{2}}$$

$$+ 2.81$$

RING SECTION	A	d	M = Ad	I_x Md	I_G
4.31 x ⅝	2.69	12.312	33.12	407.8	0.087
1½" x 12	18	6	108	648	216
	20.69		141.12	1271.8	

$$I_{NA} = I_x - \frac{M^2}{A}$$

$$= 1271.8 - \frac{141.12^2}{20.69}$$

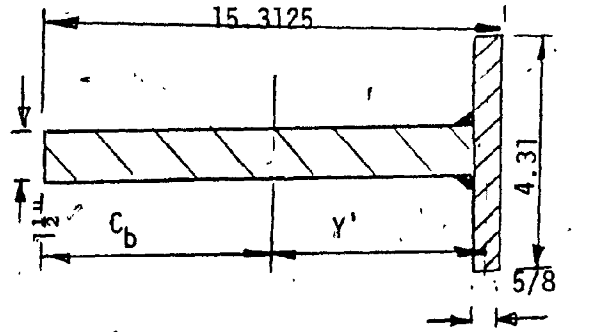
$$= 309.26 \text{ in}^4$$

$$NA\ C_b = \frac{M}{A} = \frac{141.12}{20.69}$$

$$= 6.82 \text{ in}$$

$$y' = 12.3125 - 6.82 = 5.493 \text{ in}$$

LOWER SUPPORT RING



$$e = 2.81$$

RING SECTION	A	d	M = Ad	I_x Md	I_G
4.31 x ⅝	2.69	15.3125	41.19	630.7	0.087
1½" x 15	22.5	7.5	168.75	1265.6	421.8
	25.19		209.9	2318.2	

$$I_{NA} = I_x - \frac{M^2}{A}$$

$$= 2318.2 - \frac{209.9^2}{25.19}$$

$$= 569.17 \text{ in}^4$$

$$NA\ C_b = \frac{209.9}{25.19}$$

$$= 8.33 \text{ in}$$

$$Y' = 15.3125 - 8.33 = 6.983$$

Table 4.5: Properties of Support Rings

Table 4.6

Factors for Stresses in Support Rings
(Stresses at Half-Way Points Do Not Govern this Design)

	Formula For Tangential Tensile Force $T = K_1\ f_1$		Formula For Bending Moment M_r in Ring $M_r = K_2\ f_1\ r_c$	
FOR EIGHT POINT SUPPORT	Values for K_1		Values for K_2	
	At Hangers	Half-way Between Hangers	At Hangers	Half-way Between Hangers
	1.207	1.306	+0.065	-0.033
	Resulting Tensile Stress $\sigma_{ct} = \frac{T}{A}$		Resulting Bending Stress $\sigma_{cb} = \frac{M_r}{S}$	

Now consider the tangential forces on support rings:

$$f_1 = \frac{42 \times 12}{61.5} = 8.20 \text{ KIP} \quad \text{(see Fig. 4.3)}$$

Upper Ring $T_T = K_1\ f_1 = 1.207 \times 8.20 = 9.90$ KIP

$$\sigma_{c_T} = \frac{T_T}{A_T} = \frac{9.90}{20.69} = 0.48 \text{ KSI}$$

Lower Ring $T_B = K_1(f_1 + 15 + \frac{24.860}{4})$

$$= 1.207\ (8.20 + 15 + 6.215) = 35.50 \text{ K}$$

$$\sigma_{cB} = \frac{T_B}{A_B} = \frac{35.5}{25.19} = 1.41 \text{ Ksi}$$

$$M_r = K_2\ f_1\ r_c$$

$$= 0.065 \times 8.20 \times 63$$

$$= 33.58 \text{ KIP in}$$

$$\sigma_{cbT} = \frac{M_r y}{I_T} = \frac{33.58 \times 6.82}{309.26} = 0.74 \text{ KSI}$$

$$\sigma_{cbB} = \frac{M_r y}{I_B} = \frac{0.065 \times 35.5 \times 63 \times 8.33}{569.17} = 2.13 \text{ KSI}$$

$$\text{Pressure Stress} = \sigma_{cp} = \frac{P_r}{t} \times \frac{\text{area of shell in section}}{\text{area of ring section}}$$

$$= 17.5 \times \frac{2.69}{20.69}$$

$$2.28 \text{ KSI}$$

Stresses due to pressure, exchanger weight and imposed lateral shear loads:

$$\sigma_{TOTAL} = \sigma_{cp} + \sigma_{cT} + \sigma_{cb}$$

$$\text{Upper Ring } \sigma = 2.28 + 0.48 + 0.74$$

$$= 3.50 \text{ KSI} < S$$

$$\text{Lower Ring } \sigma = 2.28 + 1.41 + 2.13$$

$$= 5.82 \text{ KSI} < S$$

The above stresses are due to pressure, maximum operating weight and imposed lateral loads only.

B. Stresses Due to External Moments

We will now investigate stresses due to externally applied loads.

Calculate combined reactions:

$I_{xx} = I_{yy} = 14400$ Unit in^2

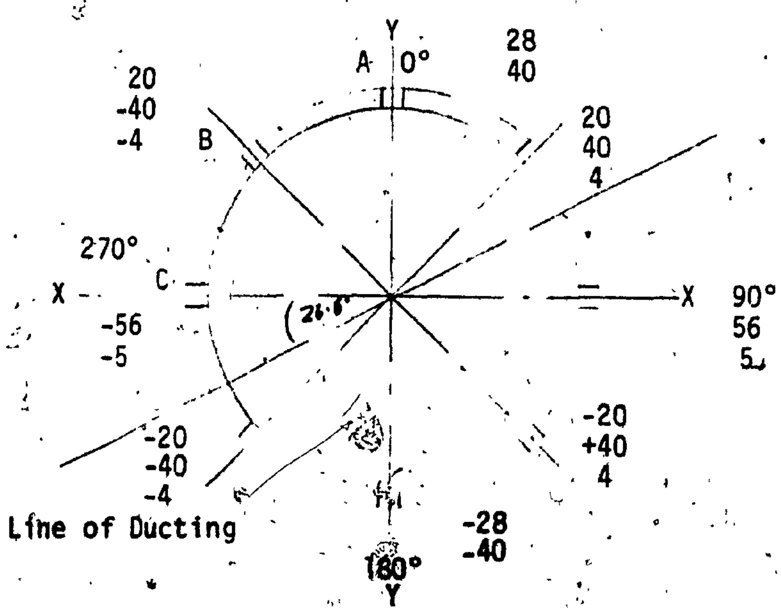
Section Modulus $= \frac{14400}{60} = 240$ Unit in

External moment from ducting = 15 000 KIP in

$M_{yy} = 15\ 000 \cos 26.6 = 13\ 413$ KIP in

$M_{xx} = 15\ 000 \sin 26.6 = 6717$ KIP in

Fig. 4.5: Reactions Due to Moments From Ducting:



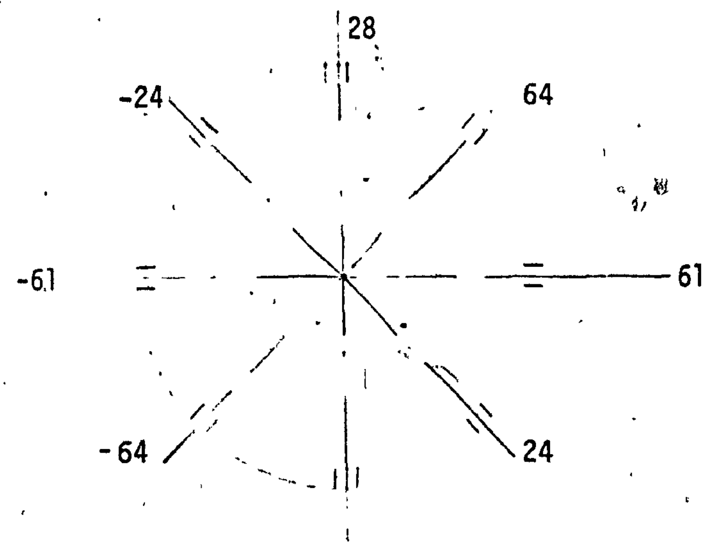


Fig. 4.6: Reactions Due to External Moments Only

M_{xx}	M_{yy}
$R_A = \pm \frac{6717}{240} = \pm 28$ KIP	$R_A = 0$
$R_B = \frac{6717}{14400} \times \frac{60}{\sqrt{2}} = \pm 20$ KIP	$R_B = \frac{13413}{14400} \times \frac{60}{\sqrt{2}} = \pm 40$ K
$R_C = 0$	$R_C = \frac{13413}{240} = \pm 56$ K

Reactions due to moments from piping:

$M_{yy} = 1080$ KIP in

$R_A = 0$

$R_B = \pm \frac{1080}{14400} \times \frac{60}{\sqrt{2}} = 3.2$ K

$R_C = \frac{1080}{240} = 4.5$

Reactions due to lowest operating weights of 226 KIP.

$$R_A = R_B = R_C = \frac{226}{8} = 28 \text{ KIP}$$

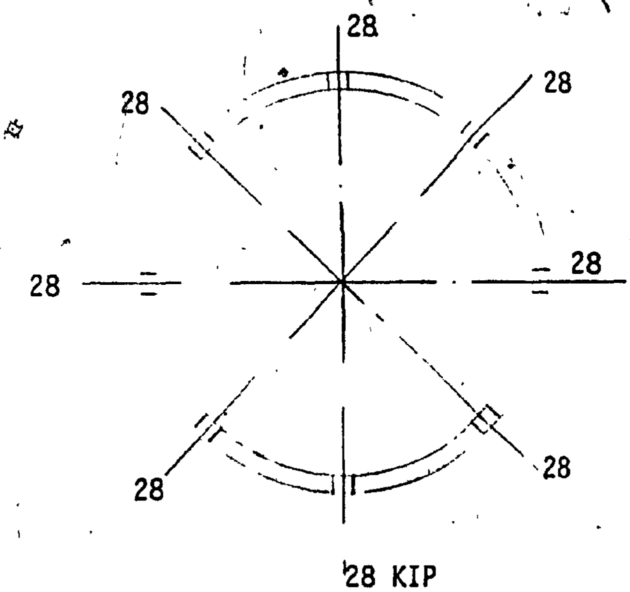


Fig. 4.7:

Reactions due to higher operating weights of 321 KIP + say 12 KIP insultation = 333 KIP

$$R_A = R_B = R_C = \frac{333}{8} = 42 \text{ KIP}$$

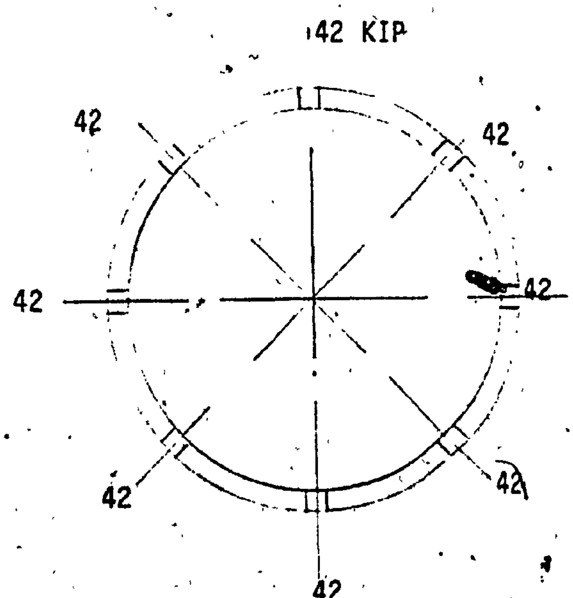


Fig. 4.8:

Following are combined reactions:

To determine maximum uplift, the overturning reactions are combined with the lowest operating weight reactions.

To determine the maximum positive reactions, the reactions due to overturning are combined with the higher operating weight reactions.

Total WHE Wind Shear = 14 KIP

Total WHE Earthquake Shear. = 40 KIP

Shear From Ducting = 25 KIP

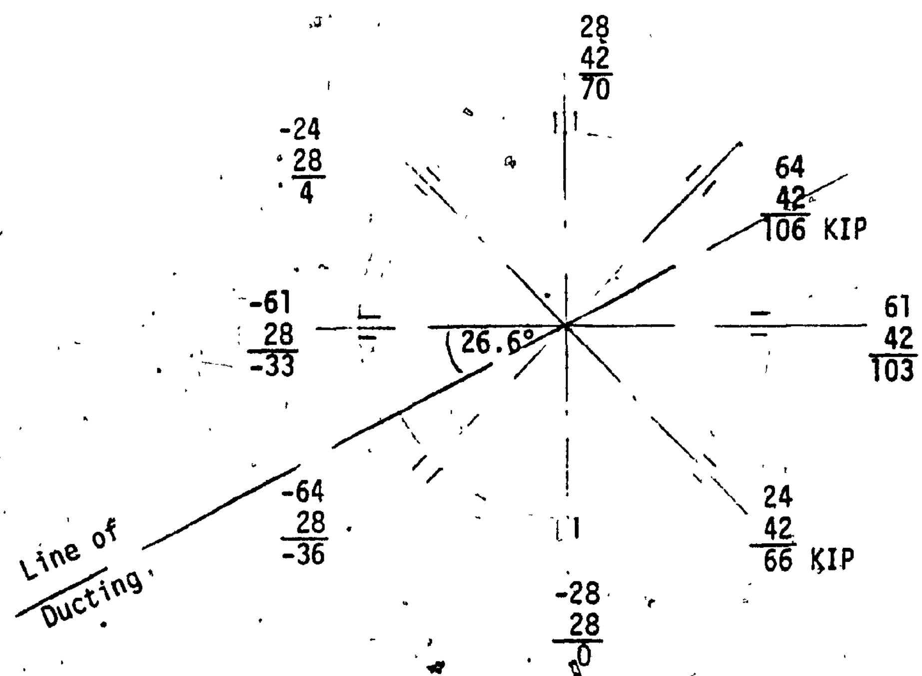


Fig. 4.9: Combined Reactions in KIPS

Others - To design the supporting structural steel for the above reactions including uplifts, friction etc. If it is not feasible to design for uplifts then the overturning moment must be reduced. Other combinations of directions of externally applied moments should also be investigated.

Loading to be considred for sizing of anchor bolts:

Wind Shear on Exchanger = 14 KIP

Lateral Earthquake

Shear on Exchanger = 0.12 x 333 = 40 KIP

Ducting imposed loads:

Lateral Shear = 25 KIP

Say 30 KIP maximum

Moment = 15 000 KIP in

Maximum Uplift = 32 KIP

Downcomer Moment = 6 x 1 x 15 x 12 = 1080 KIP in

This is about M_{yy}. With downcomers at 15' - 0"

$R_A = 0$ $\qquad R_B = \pm \dfrac{1080}{14400} \times \dfrac{60}{\sqrt{2}} = \pm 3.2$ KIP

$$R_C = \pm \frac{1080}{240} = \pm 4.5 \text{ KIP}$$

$\therefore$ combined maximum uplift from ducting and piping =

32 + 4 = 36 KIP

Say 40 KIP per support point.

Use 1⅜" DIA bolts to SA 193 GR B7

1¼" corroded DIA area = 0.929 in²

Tensile stress due to maximum uplift $= \dfrac{40}{2 \times 0.929} = 21.5$ KSI OK

Maximum shear stress:

Shear load = 40 + 30 = 70 KIP Total

Say 8 bolts are effective on 4 supports $\tau = \dfrac{70}{8 \times 0.929} = 9.42$ KSI OK

To check localized stresses near bolts in the lower ring plate due to uplift use, "Engineering Monograph 27" Moments and Reactions for Rectangular Plates. A water resources technical publication - U.S. Department of Interior Bureau of Reclamation [23].

$$\frac{a_p}{b_p} = \frac{7.5}{15} = 0.5$$

SAY $a_p = 7.5"$

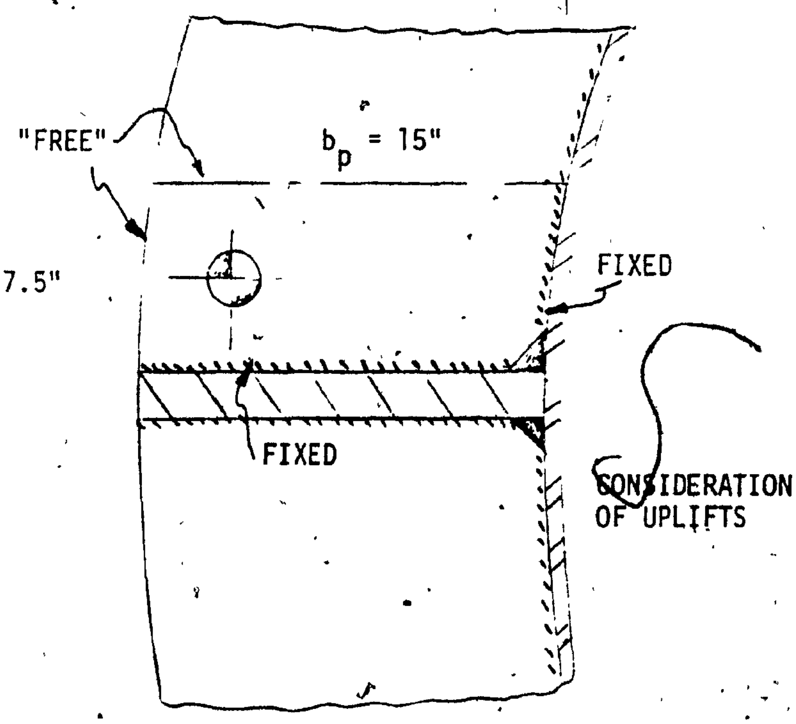


Fig. 4.10

Very conservative load = 40 KIP $\quad p = \frac{40}{15 \times 7.5} \times 144 = 51 \frac{KIP}{ft^2}$

Moment coefficient = $Pb^2 = 51.0 \times 1.25^2 = 80$ KIP

$$\sigma_b = \frac{M}{Z} \qquad Z = \frac{t^2}{6}$$

$$M_x = 0.1074 \times 80 = 8.59 \frac{KIP\ ft}{ft} \qquad M_y = 0.105 \times 80 = 8.41 \frac{KIP}{ft}\ ft$$

$$\sigma = \frac{6M}{t^2} = \frac{6 \times 8.59}{1.5^2} = 22.42 \text{ KSI}$$

$$< 1.5S = 1.5 \times 17.5$$

$$= 26.25 \text{ KSI}$$

OK

Alternatively consider the ring plate as a cantilever with point load.

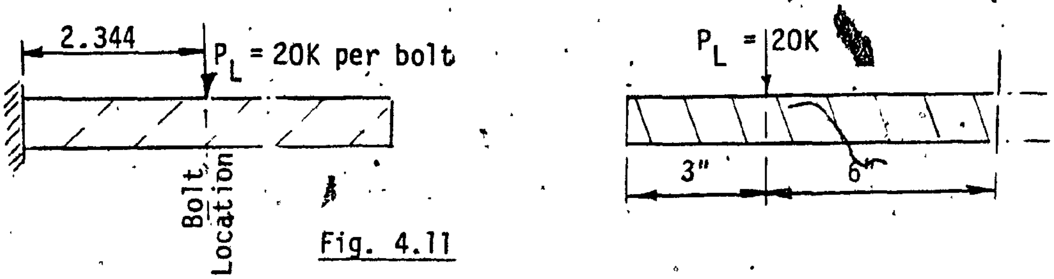


Fig. 4.11

Say $I = \frac{9 \times 1.5^3}{12} = 2.531 \text{ in}^4$

$M = 2.344 \times 20 = 46.875$ KIP in

$\sigma = \frac{M}{Z} = \frac{46.875}{2.531} \times 0.75 = 13.89$ KSI

Check fillet welds

Say with allowable $\sigma = 12.4$ KSI

$\frac{1}{16}$ FW can take 0.55 KIP/in

$\frac{5}{16}$ FW can take 2.75 KIP/in

Capacity in 9" of $\frac{5}{16}$" weld = 9 x 2.75 = 24.75 KIP

= 24.75 > 20 KIP OK

Check fillet weld capacity of supports:

Stiffeners

Shear Capacity = 2 x 60 x 6 x 0.55 conservative

F_V = 330 KIP each OK

Stiffener to bottom ring weld

F_v = 2 x 15 x 5 x 0.55 = 82.5 KIP each OK

Ring to shell weld local to supports - say 18 in effective

F_v = 2 x 18 x 5 x 0.55 = 99 KIP OK

$\frac{5}{16}$ in fillet welds OK

Now consider the effects of external moments on the WHE shell. From Fig. 4.6 the maximum R_M generated by external moment is R_M = 64 KIP.

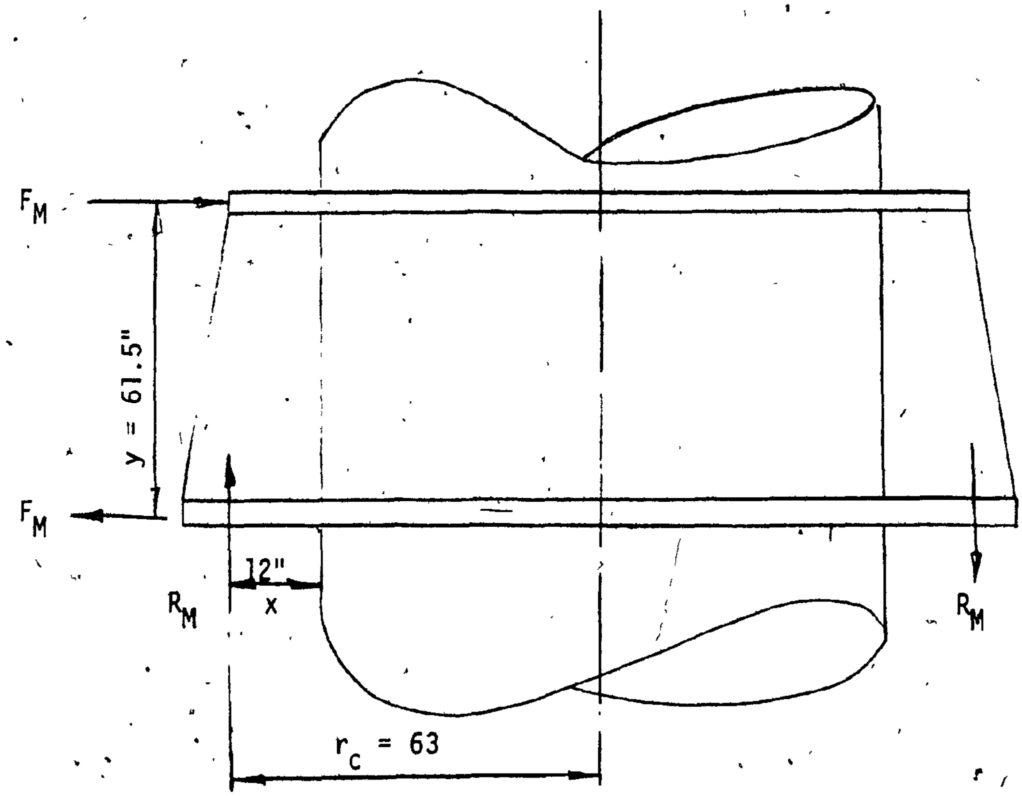


Fig. 4.12: Effect of External Moment on Support Ring

$$F_M = \frac{R_M\ x}{y} = \frac{64 \times 12}{61.5} = 12.5 \text{ KIP}$$

Then using Roark Table VIII load cases 24, 5 and 1 [28], the maximum moment at ring is

$$M = 0.1593\ F_M\ r_c$$
$$= 0.1593 \times 12.5 \times 63$$
$$= 125.5 \text{ KIP in}$$

$$\sigma_{cb\ T} = \frac{M}{I} y = \frac{125.5}{209.26} \times 6.82 = 2.77 \text{ KSI}$$

$$\sigma_{cb\ T} = \frac{125.5}{569.71} \times 8.33 = 1.83 \text{ KSI}$$

C. Combined Stresses Due to Maximum Operating Weight and Overturning Moment

Upper Ring σ = 3.5 + 2.77 = 6.27 KSI

Lower Ring σ = 5.82 + 1.83 = 7.65 KSI

Lower Ring on Uplift Side σ = 5.82 + 13.89

= 19.71 KSI

OK

4.4 CONCLUSIONS

It has been shown that the waste heat exchanger has been designed in accordance with the rules of ASME Section VIII, Division 1.

CHAPTER 5

CHAPTER 5

DESIGN AND ANALYSIS OF TUBEPLATES AND TUBE TO TUBEPLATE JOINTS

5.1 TUBE TENSION DUE TO PRESSURE

The tube tension will be calculated accurately using a computer code, in a later section. As a first estimate, this will be obtained by hand calculation. The loads induced in the tube by pressure are as follows:

1. Tube Side Pressure

$$P_T = P_D A = 40 \times \frac{\pi}{4} (2.5 - 0.4)^2 = 138.5 \text{ lbf}$$

2. Shell Side Pressure

Assuming zero shell contribution

$$P_S = \frac{200}{470} \left(\frac{\pi}{4}\ 100.5^2 - 470 \times \frac{\pi}{4} \times 2.5^2\right)$$

$$= \frac{200}{470} (7932.72 - 2307.11)$$

$$= 2394 \text{ lbf}$$

3. Total Tube Force $F = P_T + P_S$

$$= 1385 + 2394$$

$$= 2533 \text{ lbf}$$

$$\text{Tube Stress} = \frac{F}{A_t} = \frac{2533}{1.45} = 1746 \text{ psi}$$

This compares well with the value obtained from maxium tube tension using Ansys computer code - see later.

5.2 ALLOWABLE LOADS FOR TUBE TO TUBE PLATE JOINTS

Using Appendix A of the ASME, Section VIII, Division 1, Para. UA 002, for joint type g which is rolled, single groove and welded with $a < 1.4\ t$

$$L_{max} = A_t\ S_a\ f_e\ f_r\ f_y$$

where

L_{max} = maximum allowable axial load in either direction in pounds

A_t = nominal transverse cross sectional area of tube wall = 1.45 in²

S_a = code allowable stress in tension of tube material (SA 192) at temperature = 11 900 psi

f_e = factor for the length of the roller expanded position of tube $\frac{\ell}{d_o}$

ℓ = length of the expanded portion of the tube = 1.4375 in

d_o = tube outside diameter = 2.5 in

$$f_e = \frac{\ell}{d_o} = \frac{1.4375}{2.5} = 0.575$$ (This is very conservative f_e = 1 is acceptable)

f_r = factor for reliability of joint = 0.65

f_y = factor for difference in the mechanical properties of tubeplate and tube material = 1.0

$$L_{max} = A_t\ S_a\ f_e\ f_r\ f_y$$

$$= 1.45 \times 11\ 900 \times 0.575 \times 0.65 \times 1$$

$$= 6449\ \text{lbf}$$

$$L_{max} > P = 2533\ \text{lbf}$$

5.3 ALLOWABLE TUBE COMPRESSIVE STRESS

Using standards of Tubular Exchanger Manufacturers Association, TEMA, the allowable tube compressive stress, psi, for the tubes at the periphery of the bundle is given by

$$S_c = \frac{\pi^2 E_t}{2\left(\frac{K\ell}{r}\right)^2} \qquad \text{when} \quad C_c < \frac{K\ell}{r}$$

where

$$C_c = \sqrt{\frac{2\pi^2 E_t}{S_Y}}$$

S_Y = yield stress of tube material

= 26 000 psi

r = radius of gyration of tube

= 0.8149 in

$K\ell$ = equivalent unsupported buckling length of the tube in.

ℓ = unsupported tube span inch

K = 0.8 for unsupported spans between tube plate and support plate

K = 1.0 for unsupported spans between two support plates

E_t = elastic modulus of tube material at mean metal temperaure

= 28.3×10^6 psi

For span between support plate and tube plate

$$\frac{K\ell}{r} = \frac{0.8 \times 225}{0.8149} = 220.89 \quad \text{maximum}$$

For span between two support plates

$$\frac{K\ell}{r} = \frac{1.0 \times 180}{0.8149} = 220.89$$

$$C_c = \sqrt{\frac{2\pi^2 \times 28.3 \times 10^6}{26\ 000}}$$

$$= 146.58$$

$$\therefore \quad C_c < \frac{K\ell}{r}$$

$$S_c = \frac{\pi^2 E_T}{2\left(\frac{K\ell}{r}\right)^2} = \frac{\pi^2 \times 28.3 \times 10^6}{2 \times 220.89^2}$$

$$= 2862 \text{ psi}$$

5.4 CALCULATION OF TUBE AND TUBE PLATE STRESSES

The tube plate was designed using the stayed plate method in which the end pressure force is taken largely by the tubes.

The tube and tube plate stresses were calculated using Ansys computer code using method similar to that outlined in the paper "Calculation of Tube Plates in Heat Exchangers" by B. Barp and R. Angehrn [17].

The tube plate shown in Fig. 5.1 is subdivided into circular ring shaped computation elements (Fig. 5.2). Each circular ring element has a stiffness dependent on its thickness and its perforations and is regarded as a fully axisymmetric element.

The continuous flexible support of the plate provided by the tubes is simulated by annular supports at each interconnection line of the circular ring shaped computation elements. The logitudinal and bending

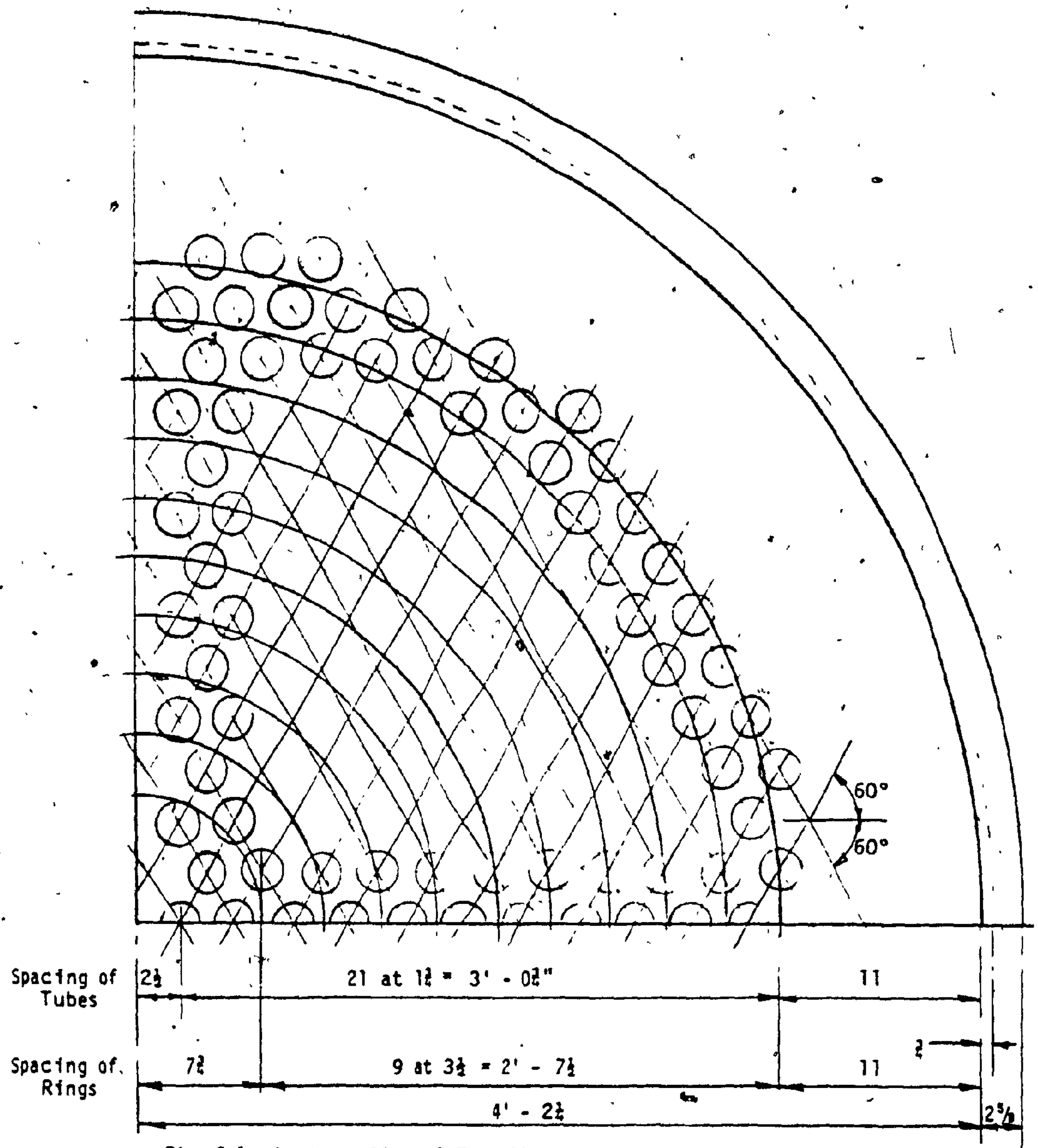


Fig. 5.1 Quadrant Plan of Tube Plate - Showing Modelling of Tubes by Equivalent Rings.

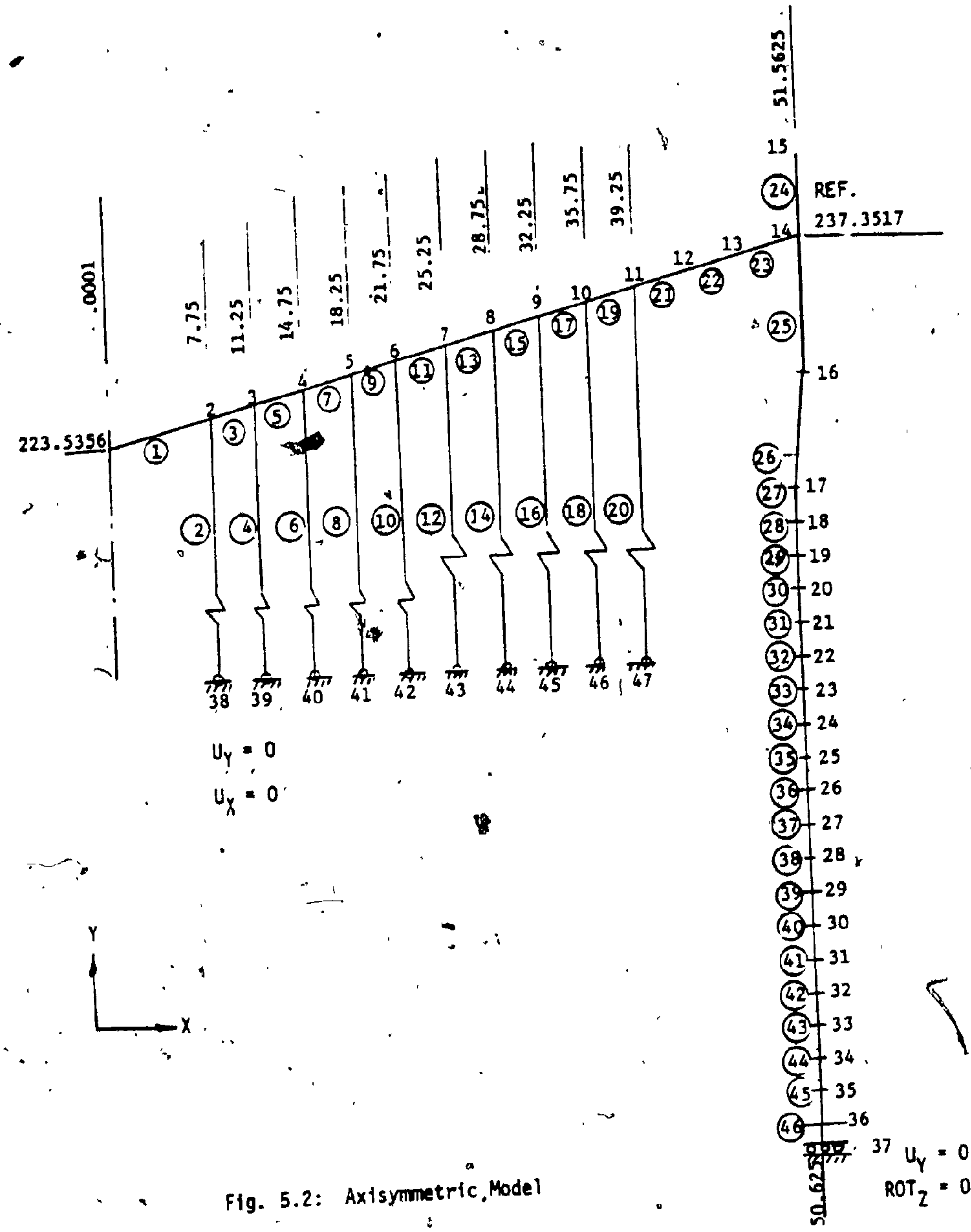


Fig. 5.2: Axisymmetric Model

stiffnesses of these supports represent the sums of the corresponding stiffnesses of the tubes assigned to this region in the computation. Although the supports have a bending stiffness in the radial direction, they have no stiffness in the tangential direction, so that together they form a tube of orthotropic material without stiffness in its circumferential direction.

It is assumed that both tube plates are identically designed. This is conservative.

The pressure differences and also temperature differentials, occuring between the individual parts, were used as load parameters.

<u>Ansys Run Tube</u> —— Axisymmetric Model

Nodes and elements numbering and boundary conditions are as shown in Fig. 5.2.

Node coordinates are as shown on Table 5.2.

Element Types : Tube Plate - STIFF 11

Shell - STIFF 11

Tubes - STIFF 14

E^* & ν^* for perforated part of tube sheet obtained from p. 388, Article 4.9, ASME, Section VIII, Div. 2. For

$t = 1\frac{1}{2}"$ $\quad p = 3\frac{1}{2}"$ $\quad$ DIA of hole, $D = 2\frac{1}{2}"$

$h = P - D = 1$

$$\frac{t}{P} = \frac{1.5}{3.5} = .429$$

$$\frac{h}{P} = \frac{1}{3.5} = .286$$

$$\frac{E^*}{E} = .233 \quad \text{(By Interpolation)}$$

$E = 2.713 \times 10^7$ psi

$$\begin{cases} E^* = 6.3212 \times 10^6 \text{ psi} \\ \nu^* = .395 \quad \text{(By Interpolation)} \end{cases}$$

E & ν for solid part of tube plate and shell.

$$\begin{cases} E = 2.713 \times 10^7 \text{ psi} \\ \nu = .3 \end{cases}$$

Density $= \dfrac{.283565 \text{ PCI}}{G} = 7.345 \times 10^{-4}$ ($G = 386.088$ in/sec^2)

Coefficient of thermal expansion - 6.75×10^{-6}/°F

Ref. Temperature (Arbitary) 0°F

Temperature of tubes 25°F for computer input, the actual design will be for ΔT = 45°F

Temperature of shell and tube plate 0°F

Pressure 200 psig

Thickness of shell : as shown on ~~JSK1/565-981~~ and Fig. 5.3

Thickness of tube plate : as shown on JSK1/565-981 : 1½"

Equivalent areas and I_{zz} for tubes: (tubes as shown on JSK1/565-981)and Fig.5.1

$$\text{Area/Tube} = A_t = \frac{\pi \times 2.5^2}{4} - \frac{\pi\,(2.5 - .2 \times 2)^2}{4} = 1.44513262 \text{ in}^2$$

$$I_{zz}\text{/Tube} = I = \frac{\pi \times 2.5^4}{64} - \frac{\pi\,(2.5 - .2 \times 2)^4}{64} = .9628196085 \text{ in}^4$$

Depth of Tube = 2½"

RADIUS	CIRCUMFERENCE	= NO. OF TUBES	$= \frac{NA}{2\pi} \frac{AREA}{RADIAN}$	$= \frac{NI}{2\pi} \frac{I_{zz}}{RADIAN}$
7.75"	π x 15.5	15.5	3.565	2.3752
11.25	π x 22.5	22.5	5.175	3.4478
14.75	π x 29.5	29.5	6.785	4.5205
18.25	π x 36.5	36.5	8.395	5.5932
21.75	π x 43.5	43.5	10.005	6.6658
25.25	π x 50.5	50.5	11.615	7.7385
28.75	π x 57.5	57.5	13.225	8.8112
32.25	π x 64.5	64.5	14.835	9.8838
35.75	π x 71.5	71.5	16.445	10.9565
39.25	π x 78.5	78.5	18.005	12.0291
TOTAL	π x 470	470.	$\frac{679.21}{2\pi}$	$\frac{452.53}{2\pi}$

Table 5.1: Distribution of the Tubes Over the Interconnection Lines Between the Individual Circular Ring-Shaped Computation Elements.

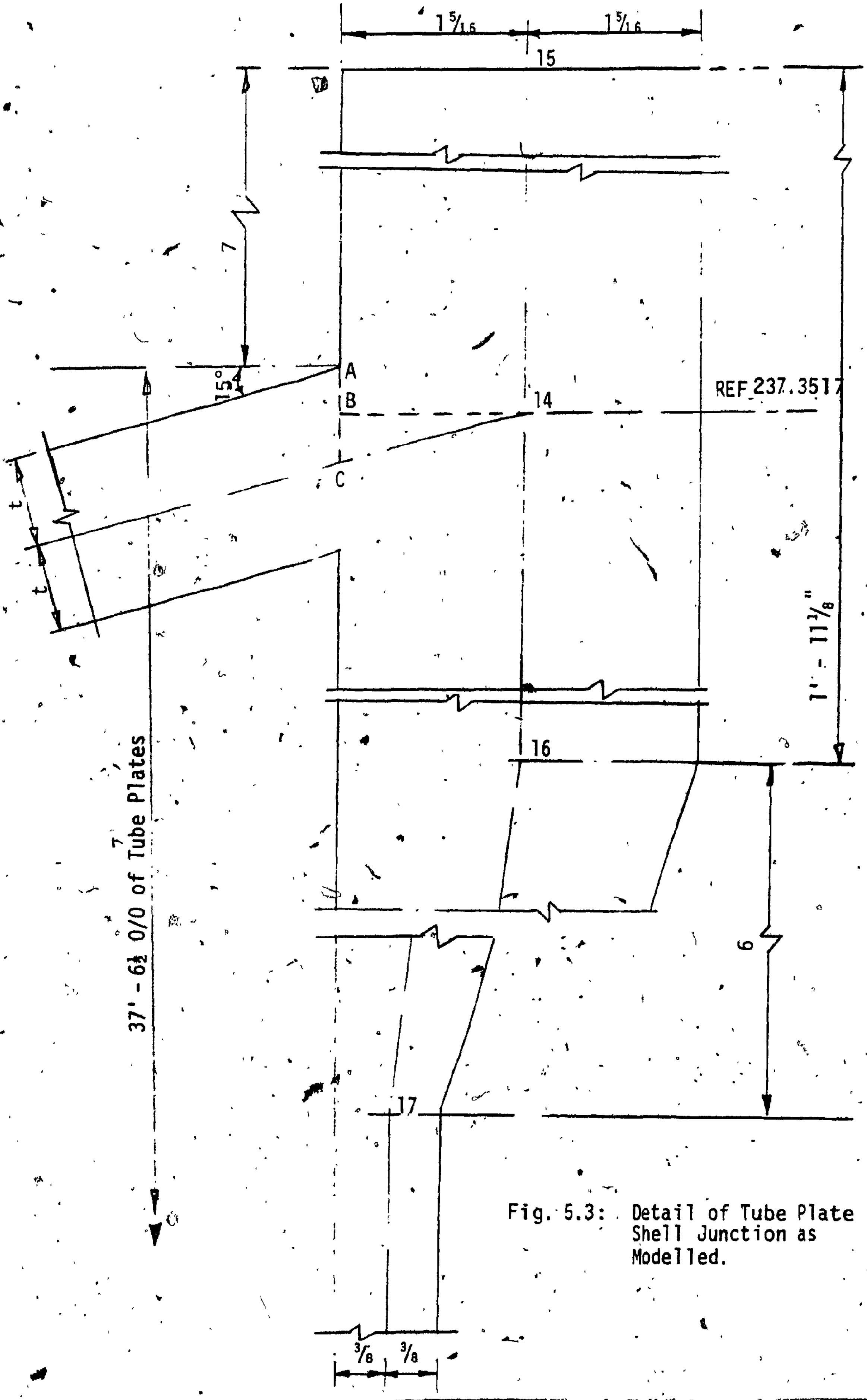


Fig. 5.3: Detail of Tube Plate Shell Junction as Modelled.

SOME NODE COORDINATES FOR FIG. 5.2

NODE NUMBER	X	ELEVATION Y
1	0.001	223.5355 799
2	7.75	225.6121 593
3	11.25	226.5499 815
4	14.75	227.4878 037
5	18.25	228.4256 258
6	21.75	229.3634 48
7	25.25	230.3012 702
8	28.75	231.2390 924
9	32.75	232.1769 145
10	35.75	233.1147 367
11	39.25	234.0525 589
12	42.25	234.8564 065
13	44.4233	235.4387 404
14	51.5625	237.3516 833

TABLE 5.2

Table 5.3: Spring Constants

x	ELEMENT#	N	1¼" TUBE PLATE FULL TUBE	1¼" TUBE PLATE CORRODED TUBE	1½" TUBE PLATE FULL TUBE	1½" TUBE PLATE CORRODED TUBE
7.75	2	15.5	441466.	174547.	441727.	174650.
11.25	4	22.5	639469.	252833.	639846.	252982.
14.75	6	29.5	836628.	330786.	837120.	330981.
18.25	8	36.5	1032948.	408407.	1033555.	408647.
21.75	10	43.5	1228435.	485699.	1229155.	485984.
25.75	12	50.5	1423094.	562664.	1423926.	562993.
28.75	14	57.5	1616931.	639303.	1617874.	639676.
32.25	16	64.5	1809950.	715619.	1811004.	716035.
35.75	18	71.5	2002156.	791613.	2003320.	792073.
39.25	20	78.5	2193556.	867289.	2194828.	867792.

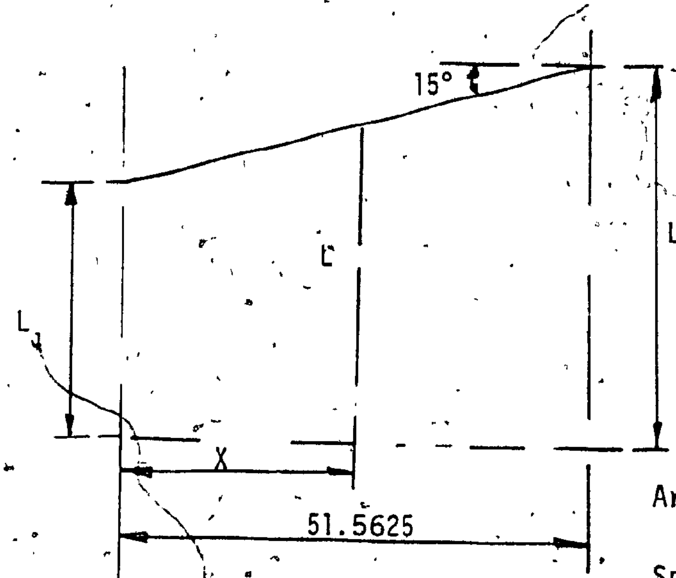


Fig. 5.4: Node Coordinates

$$L_0 = Y_{14} - Y_{37}$$

$$L_1 = L_0 - (51.5625 \tan 15°)/2$$

$$L = L_0 - (51.5625 - x) \tan 15°/2$$

$$\frac{\text{AREA}}{\text{RADIAN}} = \frac{NA}{2\pi} = \frac{xA}{\pi}$$

$$\text{Area (full tube)} = \frac{\pi \times 2.5}{4} - \frac{\pi \times 2.1^2}{4}$$

$$\text{Area (corroded tube)} = \frac{\pi \times 2.5}{4} - \frac{\pi \times 2.35^2}{4}$$

$$\text{Spring Constant} = \frac{\frac{\text{AREA}}{\text{RADIAN}} \times E}{L}$$

L is correct for tube plate full thickness and not modified for corroded tube plate thickness.

KEY TO ANSYS COMPUTER RUNS

Tubes Represented by Springs

TUBE 21 1¼ tube plate
full tube plate/shell
full tubes

TUBE 22 1¼ tube plate
full tube plate/shell
corroded tubes

TUBE 23 1¼ tube plate
corroded tube plate/shell
corroded tubes

TUBE 31 1½ tube plate
full tube plate/shell
full tubes

TUBE 32 1½ tube plate
full tube plate/shell
corroded tubes

TUBE 33 1½ tube plate
corroded tube plate/shell
corroded tubes

TABLE 5.4: Summary of Output From Ansys Runs

FILE NAME / DESCRIPTION	LOADING	MAX. TUBE STRESS (KSI) (AXIAL)	MAX./MIN. TUBE SHEET STRESS (KSI)		MAX. SHELL HOOP STRESS (KSI)
			FACE 1	FACE 2	
TUBE 21 ALL PARTS UNCORRODED	ΔT = 25°F	- .883	+ 7.023	- 9.048	-*
	200 psig	+1.740	+23.318	-23.829	+14.359
TUBE 22 TUBES ONLY CORRODED 1/8"	ΔT = 25°F	-1.373	+ 5.704	- 7.567	*
	200 psig	+3.641	+26.854	-28.358	+14.363
TUBE 23 ALL PARTS CORRODED 1/8"	ΔT = 25°F	-1.236	+ 5.457	- 7.382	*
	200 psig	+3.927	+30.143	-31.843	+17.350

Notes: ΔT = Temp. of Tubes - Temp. of Tube Plate/Shell

Tube Plate $1\frac{1}{4}$" (Uniform Uncorroded Thickness)

$$\text{Stress in Tube} = \frac{2\pi}{NA} \times (\text{Force in Tube/Rad.})$$

$$= .0554 \times (\text{Force in Tube/Rad.})$$

$N = 78.5$

For Uncorroded Tube, $A = 1.445 \text{ in}^2$ } From Table 5.1

For Corroded Tube, $A = 0.5714 \text{ in}^2$

* Negligible where max. hoop stress occurs under pressure loading.

TABLE 5.5: Summary of Output From Ansys Runs

FILE DESCRIPTION	LOADING	MAX. TUBE STRESS (KSI) (AXIAL)	MAX./MIN. TUBE SHEET STRESS (KSI)		MAX. SHELL HOOP STRESS (KSI)
			FACE 1	FACE 2	
TUBE 31	$\Delta T = 25°F$	-1.026	+ 7.165	- 8.931	*
ALL PARTS UNCORRODED	200 psig	+1.450	+19.321	-19.765	+14.341
TUBE 32	$\Delta T = 25°F$	-1.548	+ 5.761	- 7.378	*
TUBES ONLY CORRODED 1/8	200 psig	+3.090	+22.466	-23.746	+14.342
TUBE 33	$\Delta T = 25°F$	-1.421	+ 5.627	- 7.277	*
ALL PARTS CORRODED 1/8	200 psig	+3.283	+24.710	-26.142	+17.311

Actual Measured Tube Plate Thickness is 1½" New.

Notes: ΔT = Temp. of Tubes - Temp. of Tube Sheet/SHE

Tube Plate 1½" (Uniform Uncorroded Thickness)

$$\text{Stress in Tube} = \frac{2\pi}{NA} \times (\text{Force in Tube/Radian})$$

$$= .0554 \times (\text{Force in Tube/Radian})$$

$$\left.\begin{array}{ll} & N = 78.5 \\ \text{For Uncorroded Tube,} & A = 1.445 \text{ in}^2 \\ \text{For Corroded Tube,} & A = 0.5714 \text{ in}^2 \end{array}\right\} \text{From Table 5.1}$$

* Negligible where max. hoop stress occurs under pressure loading.

5.5 COMPARISON OF CALCULATED STRESSES WITH CODE ALLOWABLE VALUES

The above stresses have been calculated for tube plate thicknesses of 1½in and 1¼ in. The actually measured tube plate thickness was 1½ in.

For a temperature differential of 25°F between tubes and shell, the maximum calculated compressive stress is 1421 psi. For conservative design we will use $\Delta T = 45°F$ from Chapter 2, Section 2.3.

∴ Maximum compressive stress for $\Delta T = 45°F$

$$\cancel{19}\ \sigma = \frac{45}{25} \times 1421 = 2558 \text{ psi}$$

Allowable stress = 2862 psi OK
(From section 5.3)

The stresses in tube plate are:

1. Due to pressure: 19 765 psi new

   26 142 psi corroded

   Only a portion of this stress is primary, but conservatively even if all of this is classified as primary stress. The ASME Code allowable $= 1.5\ S_M = 1.5 \times 21\ 700$

   $= 32\ 550$ psi

2. Primary plus secondary stresses:

   These are stresses due to pressure and differential thermal expansion.

   $$= 26\,142 + 7277 \times \frac{45}{25}$$
   $$= 26\,142 + 13\,099$$
   $$= 39\,241 \text{ psi}$$

The code allowable stress range from ASME, Section VIII, Division 2 is

$$3\ S_M = 3 \times 21\,700$$
$$= 65\,100 \text{ psi}$$

5.6 CONCLUSIONS

Above calculations show that the stresses in tubes and tube plates meet the Code allowable values for the specified operating conditions.

CHAPTER 6

CHAPTER 6

DESIGN AND FLEXIBILITY ANALYSIS OF WASTE HEAT EXCHANGER PIPING

All piping associated with this waste heat exchanger was designed and fabricated in accordance with ANSI B 31.3, Petroleum Refinery Piping Code [14].

The detailed design and flexibility analysis for downcomer piping is presented below. An identical procedure was used for the riser piping.

6.1 PRESSURE DESIGN

6.1.1 Straight Pipe Under Internal Pressure

The internal pressure design thickness (t) was calculated in accordance with paragraph 304.1.2 []4] by using the following equation:

$$t = \frac{PD_o}{2(SE + Py)} + C$$

$P = 200$ psi

$D_o = 6.625$ in

$y = 0.4$

$E = 1.0$

$S = 20\,000$ psi for SA 106 GR B pipe

$C =$ corrosion allowance $= 0.125"$

therefore for 6 in pipe

$$t = \frac{200 \times 6.625}{2\ (20\,000 + 200 \times 0.4)} + 0.125$$

$$= 0.033 + 0.125$$

$$= 0.158 \text{ in}$$

Hence, use of standard wall (0.280 in nominal and 0.245 in minimum thickness) pipe is acceptable.

6.1.2 Standard Components

In accordance with Paragraph 303 [14], the use of 6 in standard wall elbows to ANSI B 16.9 and ASTM A 234 Grade WPB is also acceptable.

6.2 FLEXIBILITY ANALYSIS FOR DOWNCOMER PIPING

Schematic layout of the downcomer piping is shown in Fig. 6.1. The downcomer piping is completely fixed at 1 (waste heat exchanger shell) and 8 (steam drum shell).

6.2.1 Requirements for Analysis

During the WHE start up the temperature, T, of downcomer piping is 230°F. Hence ΔT the temperature above stress free condition is $\Delta T = 230 - 70 = 160°F$.

Thus the net expansion of the downcomers in the vertical direction is $\Delta Y_s = \alpha\ L\ \Delta T$

$$\Delta Y_s = 6.78 \times 10^{-6} \times 511.5 \times 160$$

$$= 0.5549 \text{ in}$$

During normal operating conditions the temperature of downcomer is 370°F

$$\therefore\ \Delta T = 300°F$$

During normal operation the anchor point 1 is displaced downward, a distance ΔY_1, by the thermal expansion of the WHE shell.

$$\Delta Y_1 = \alpha\ L\ \Delta T$$

$$= 6.78 \times 10^{-6} \times 236.75 \times (370 - 70)$$

$$= -0.481 \text{ in}$$

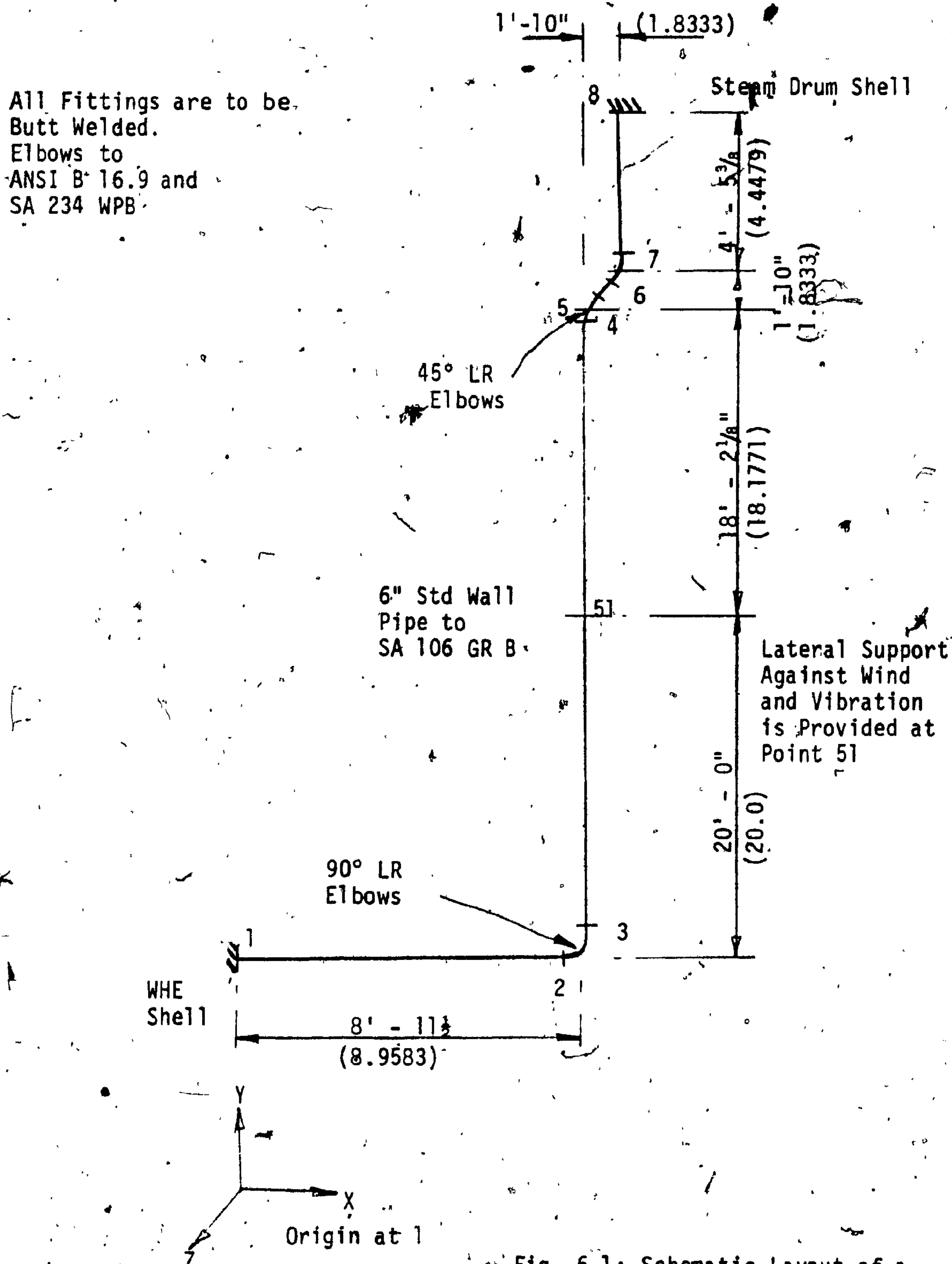


Fig. 6.1: Schematic Layout of a Downcomer

Net expansion of downcomer piping in vertical direction is

$$\Delta Y_N = (\alpha\ L\ \Delta T)_{DOWNCOMERS} - (\alpha\ L\ \Delta T)_{WHE}$$

$$= 6.78 \times 10^{-6} \times 511.5 \times 300 - 0.481$$

$$= 0.559 \text{ in}$$

Net $\Delta Y_N > \Delta Y_S$

$0.559 > 0.5549$ in

Hence, normal operating condition governs.

We will now check if analysis is mandatory in accordance with paragraph 319.4 of the code [14] using the following equation

$$\frac{DY}{(L-U)^2} \leq \frac{30\ SA}{E_A}$$

where

D = nominal pipe size, in.

Y = resultant of total displacement strains to be absorbed by the piping system, in.

U = straight line distance between anchors, feet.

L = developed length of piping between anchors, feet.

S_A = allowable stress range, psi

E_a = modulus of elasticity of the piping material in the cold condition, psi.

Expansion must be calculated for each coordinate and be combined

$$\Delta x = \alpha\ L\ \Delta T$$

$$= 6.78 \times 10^{-6} \times 129.5 \times 300$$

$$= 0.2634 \text{ in}$$

$$\Delta y = 6.78 \times 10^{-6} \times 511.5 \times 300 - 0.481 +$$

$$= 1.0404 - 0.481$$

$$= 0.559$$

$$dz = 0$$

$$Y = \sqrt{dx^2 + \delta y^2 + \delta z^2} = \sqrt{0.2634^2 + 0.559^2}$$

$$= 0.618 \text{ in}$$

$$U = \sqrt{42.625^2 + 10.7916^2} = 43.97 \text{ ft}$$

$$L = 53.417 \text{ ft}$$

$$\therefore \frac{DY}{(L-U)^2} + \frac{6 \times 0.618}{(53.417 - 43.97)^2} = 0.0415$$

$$S_A = f\,(1.25\,S_c + 0.25\,S_h)$$

$$f = \text{fatigue factor} = 1$$

$S_c = S_h = 18\ 000$ psi   Conservative. Use of 20 000 psi is permitted.

$$\therefore S_A = 1.5 \times 18\ 000$$

$$= 27\ 000 \text{ psi}$$

$$E_a = 27.4 \times 10^6$$

$$\frac{30\,S_A}{E_a} = \frac{30 \times 27\ 000}{27.4 \times 106} = 0.03$$

Hence $\frac{DY}{(L-U)^2}$ is greater than 0.03 $= \frac{30\,S_A}{E_a}$

Therefore analysis is required.

6.2.2 Analysis of Downcomer Piping

The downcomer piping was analyzed using a computer program called ADL Pipe, owned by Arthur D. Little Company.

The following load cases were considered:

1. Sustained Load

File 10 is for sustained load. This gives the longitudinal stresses due to pressure and deadweight including the weight of insultation, W.

Design Pressure = 200 psig

$W = 3.38$ lb/in

2. Occasional Load

File 11 contains the assessment of occasional loading stresses due to wind.

Design Pressure = 200 psig

$W = 3.38$ lb/in

For the assessment of wind loading, a load of 0.54 g was statically applied in the 'Z' direction.

3. Thermal Expansion Analysis

It has already been demonstrated that the normal operating condition represented most severe thermal expansion of the piping. Hence File 13 contains the analysis for the normal operating condition.

$\Delta T = 300°F$

$\alpha = 6.78 \times 10^{-6}$ in/in °F

$f = 1.0$

$S_c = 18\ 000$ psi

6.2.3 Adlpipe Inputs and Outputs

```
ADLPIPE PAGE    1                 ARTHUR D. LITTLE INC.      ADLPIPE STRESS ANALYSIS        05/03/79    15.27.01.
                      DOWNCOMERS AS SHOWN ON DRG 6 CONT.565-981 IMPERIAL OIL LTD. JS KANDOLA
                              ***********
                              VERSION - ADLPIPE (FAST) FEB 1977 REVISION 3C
                              REFER TO ADLPIPE MANUAL DATED APRIL 1977
                              ***********
 ***********************
  FEATURES OF ADLPIPE:
 ***********************
    1. ASME SECTION III , CLASS 1 STRESS ANALYSIS AND STRESS REPORT PER NB 3600
       BOTH 1972 AND 1974 (WINTER ADDENDA 1975)
    2. ASME SECTION III , CLASS 1 USAGE FACTOR CALCULATION
    3. ASME SECTION III , CLASS 2 AND 3 STRESS ANALYSIS AND STRESS REPORT PER NC 3600
       BOTH 1971 (INCLUDING WINTER 1972 ADDENDA) AND 1974
    4. ANSI B31.1, 1967 AND ANSI B31.1B, 1973 STRESS ANALYSIS AND REPORT
    5. ANSI B31.3, 1973 AND ANSI B31.4, 1973 STRESS ANALYSIS AND REPORT
    6. ISOMETRIC PLOT WITH SEQUENCE NUMBERS
    7. ISOMETRIC PLOTTING WITH DIMENSIONS
    8. PLAN AND ELEVATION DRAWINGS WITH DIMENSIONS
    9. NEW OFFSET CARD USED AS A MEMBER MODIFIER SPECIFIES WELD OFFSET, OVALITY, AND REDUCER CONE ANGLE
   10. BEAM ELEMENT
   11. RIGID BODY ELEMENT
   12. WIND LOADS
   13. SECTIONED ELBOWS
   14. COLD SPRING
   15. SKEWED BELLOWS
   16. MULTIPLE JOB CARDS
   17. LOADINGS STATIC - PRESSURE, DEADWEIGHT, THERMAL, STATIC ACCELERATION, DISPLACEMENTS, WIND
   18. LOADINGS DYNAMIC - SHOCK SPECTRAL RESPONSE ANALYSIS AND TIME HISTORY ANALYSIS
   19. LOADINGS - TIME DEPENDENT THERMAL TRANSIENT
                              FOR FURTHER INFORMATION OR COMMENT CONTACT:
                              1. YOUR TECHNICAL SERVICE REPRESENTATIVE
                              2. J. W. DINGWELL
                                 ARTHUR D. LITTLE, INC
                                 ACORN PARK
                                 CAMBRIDGE, MASS 02140
                                 TEL (617) 864-5770   TELEX 921436
```

GEOMETRY DOWNCOMERS AS SHOWN ON DRG 6 CONT.565-981 IMPERIAL OIL LTD. JS KANDOLA

ANCHOR			1	0.0	0.0	0.0			
RESTRAINT			1	1.	1.	1.	1.	1.	1.
JUNCTION			51	8.9583	20.0	0.0			
RESTRAINT			51						
ANCHOR			8	10.7917	44.4583	0.0			
RESTRAINT			8	1.	1.	1.	1.	1.	1.
SECTION									
PIPE		1	51	6.625	0.28	27.0	6.78	300.0	3.38
RUN	BW	1	2	8.9583	0.	0.			
ELBOW	BW	2	3	6.625	0.28		9.0		
RUN	BW	3	51	0.0	20.0	0.			
SECTION									
PIPE		51	8	6.625	0.28	27.0	6.78	300.0	3.38
RUN	BW	51	4	0.	18.1771	0.			
ELBOW	BW	4	5				9.0		
RUN	BW	5	6	1.8333	1.8333	0.			
ELBOW	BW	6	7				9.0		
RUN	BW	7	8	0.0	4.4479				
END									

DOWNCOMERS AS SHOWN ON DRG 6 CONT.565-981 IMPERIAL OIL LTD. J. KANDULA
DEADWEIGHT AND PRESSURE ANALYSIS FOR WHE 565-981 DOWNCOMERS

CONDITION 10
LOADS
DEADWEIGHT
PRESSURE
STRESS UNITS (LB/SQ IN)

B31.3 - 1973 SUMMARY OF 10 HIGHEST STRESSES FOR EACH EQUATION

***** SUSTAINED LOAD *****

	SEC	MEM	SEQ	POS	STRESS
1.	2	4	7	END	7146.
2.	2	4	6	BEG	5950.
3.	2	2	4	BEG	4691.
4.	2	5	7	BEG	3957.
5.	2	5	8	END	3789.
6.	2	2	5	END	3574.
7.	2	3	6	END	3369.
8.	1	1	1	BEG	2919.
9.	2	1	4	END	2838.
10.	2	3	5	BEG	2293.

DOWNCOMERS AS SHOWN ON DRG 6 CONT.565-981 IMPERIAL OIL LTD. JS KANDOLA
DEADWEIGHT AND PRESSURE AND WIND FOR WHE 565-981 DOWNCOMERS

LOADS
ACCELERATION
PRESSURE
STRESS UNITS (LB/SQ IN)

B31.3 - 1973 SUMMARY OF 10 HIGHEST STRESSES FOR EACH EQUATION

***** OCCASIONAL LOAD *****

	SEC	MEM	SEQ	POS	STRESS
1.	2	5	8	END	7001.
2.	1	1	1	BEG	6997.
3.	2	4	7	END	6520.
4.	2	4	6	BEG	4916.
5.	2	1	51	BEG	4709.
6.	1	3	51	END	4709.
7.	1	2	3	END	4445.
8.	2	5	7	BEG	4001.
9.	2	2	4	BEG	3739.
10.	2	3	6	END	3134.

DOWNCOMERS AS SHOWN ON DRG 6 CONT.565-981 IMPERIAL OIL LTD. JS KANDULA
THERMAL EXPANSION DURING NORMAL OPERATION WHE 565-981 DOWNCOMERS

LOADS
THERMAL
EXTERNAL
PRESSURE
STRESS UNITS (LB/SQ IN)

B31.3 - 1973 SUMMARY OF 10 HIGHEST STRESSES FOR EACH EQUATION

***** THERMAL RANGE *****

	SEC	MEM	SEQ	POS	STRESS
1.	1	1	1	BEG	15448.
2.	1	2	3	END	8613.
3.	1	2	2	BEG	5276.
4.	2	2	4	BEG	5124.
5.	2	2	5	END	4261.
6.	1	3	3	BEG	3881.
7.	2	4	7	END	2803.
8.	1	1	2	END	2377.
9.	2	1	4	END	2309.
10.	2	4	6	BEG	1941.

GEOMETRY STIFFEST RISER AS SHOWN ON DRG 7 CONT.565-981 IMPERIAL OIL LTD JSK

ANCHOR			1	0.0	0.0	0.0			
RESTRAINT			1	1.	1.	1.	1.	1.	1.
JUNCTION			51	1.4375	5.4169	-0.5954			
RESTRAINT			51						
ANCHOR			8	8.979	9.8882	-2.2306			
RESTRAINT			8	1.	1.	1.	1.	1.	1.
SECTION									
PIPE		1	51	6.625	0.28	27.0	6.78	300.0	3.38
RUN	BW	1	2	1.4375	0.4169	-0.5954			
ELBOW	BW	2	3	6.625	0.28		9.0		
RUN	BW	3	51	0.0	5.00	0.0			
SECTION									
PIPE		51	8	6.625	0.28	27.0	6.78	300.0	3.38
RUN	BW	51	4	0.0	4.4713	0.0			
ELBOW	BW	4	5	6.625	0.28		9.0		
RUN	BW	5	6	1.6352	0.0	-1.6352			
ELBOW	BW	6	7	6.625	0.28		9.0		
RUN	BW	7	8	5.9063	0.0	0.0			
END									

ADLPIPE PAGE 14 ARTHUR D. LITTLE INC. ADLPIPE STRESS ANALYSIS 05/03/79 14.36.41.
STIFFEST RISER AS SHOWN ON DRG 7 CONT.565-981 IMPERIAL OIL LTD JSK
DEADWEIGHT AND PRESSURE ANALYSIS FOR WHE 565-981 RISERS

CONDITION 20
LOADS
DEADWEIGHT
PRESSURE
STRESS UNITS (LB/SQ IN)

B31.3 - 1973 SUMMARY OF 10 HIGHEST STRESSES FOR EACH EQUATION

***** SUSTAINED LOAD *****

	SEC	MEM	SEQ	POS	STRESS
1.	1	1	1	BEG	2023.
2.	2	5	8	END	1581.
3.	1	2	3	END	1509.
4.	1	2	2	BEG	1508.
5.	2	2	4	BEG	1335.
6.	1	3	3	BEG	1296.
7.	2	4	7	END	1287.
8.	2	4	6	BEG	1267.
9.	1	1	2	END	1264.
10.	2	1	4	END	1186.

ADLPIPE PAGE 12 ARTHUR D. LITTLE INC. ADLPIPE STRESS ANALYSIS 05/03/79 14.40.18.

STIFFEST RISER AS SHOWN ON DRG 7 CONT.565-981 IMPERIAL OIL LTD JSK

DEADWEIGHT AND PRESSURE AND WIND FOR WHE 565-981 RISERS

LOADS
ACCELERATION
PRESSURE
STRESS UNITS (LB/SQ IN)

B31.3 - 1973 SUMMARY OF 10 HIGHEST STRESSES FOR EACH EQUATION

***** OCCASIONAL LOAD *****

	SEC	MEM	SEQ	POS	STRESS
1.	2	5	8	END	2432.
2.	1	2	3	END	1674.
3.	1	3	3	BEG	1380.
4.	2	2	5	END	1374.
5.	2	2	4	BEG	1310.
6.	1	1	1	BEG	1310.
7.	1	2	2	BEG	1236.
8.	2	4	6	BEG	1224.
9.	2	3	5	BEG	1222.
10.	2	1	51	BEG	1216.

STIFFEST RISER AS SHOWN ON DRG 7 CONT.565-981 IMPERIAL OIL LTD JSK
THERMAL EXPANSION DURING NORMAL OPERATION WHE 565-981 RISERS

LOADS
THERMAL
EXTERNAL
PRESSURE
STRESS UNITS (LB/SQ IN)

B31.3 - 1973 SUMMARY OF 10 HIGHEST STRESSES FOR EACH EQUATION

***** THERMAL RANGE *****

	SEC	MEM	SEQ	POS	STRESS
1.	2	5	8	END	24531.
2.	2	2	4	BEG	19376.
3.	2	2	5	END	13621.
4.	2	1	4	END	8983.
5.	1	2	3	END	7598.
6.	2	3	5	BEG	7212.
7.	2	4	6	BEG	4655.
8.	1	2	2	BEG	4640.
9.	2	4	7	END	4537.
10.	2	3	6	END	4332.

ADLPIPE PAGE 20 ARTHUR D. LITTLE INC. ADLPIPE STRESS ANALYSIS 05/03/79 14.47.14.
STIFFEST RISER AS SHOWN ON DRG 7 CONT.565-981 IMPERIAL OIL LTD JSK
STRESS REPORT FOR RISERS WHE 565-981
STRESS UNITS (LB/SQ IN)

B31.3 - 1973 SUMMARY OF 10 HIGHEST STRESSES FOR EACH EQUATION

***** SUSTAINED LOAD *****

	SEC	MEM	SEQ	POS	STRESS
1.	1	1	1	BEG	2023.
2.	2	5	8	END	1581.
3.	1	2	3	END	1509.
4.	1	2	2	BEG	1508.
5.	2	2	4	BEG	1335.
6.	1	3	3	BEG	1296.
7.	2	4	7	END	1287.
8.	2	4	6	BEG	1267.
9.	1	1	2	END	1264.
10.	2	1	4	END	1186.

***** OCCASIONAL LOAD *****

	SEC	MEM	SEQ	POS	STRESS
1.	2	5	8	END	2419.
2.	1	1	1	BEG	1971.
3.	1	2	3	END	1904.
4.	2	2	4	BEG	1584.
5.	1	3	3	BEG	1520.
6.	2	2	5	END	1358.
7.	2	4	7	END	1347.
8.	1	2	2	BEG	1333.
9.	2	1	4	END	1309.
10.	1	3	51	END	1297.

***** THERMAL RANGE *****

	SEC	MEM	SEQ	POS	STRESS
1.	2	5	8	END	24531.
2.	2	2	4	BEG	19376.
3.	2	2	5	END	13621.
4.	2	1	4	END	8983.
5.	1	2	3	END	7598.
6.	2	3	5	BEG	7212.
7.	2	4	6	BEG	4655.
8.	1	2	2	BEG	4640.
9.	2	4	7	END	4537.
10.	2	3	6	END	4332.

S_h = 18 000 psi

ΔY_1 = -0.481 in = imposed displacement at anchor Point 1.

Boundary Conditions

In all load cases the anchor points 1 and 8 were completely fixed. In addition, for load File 13, the anchor point 1 was displaced ΔY_1 = - 0.481 in.

The results of the analysis are shown on pages 95-99.

6.3 ANALYSIS OF RISER PIPING

The 6 in. riser piping was analyzed using an identical approach to the downcomer. An isometric sketch of a typical riser loop is shown in Fig. 6.2.

Adlpipe input for riser piping is shown on pages 100-104. These calculations show that the stresses in riser piping for the specified operating conditions are within the code allowable values.

6.4 CONCLUSIONS FROM PIPING ANALYSIS

These flexibility calculations show that the downcomer and riser piping meet the ANSI B 31.3 Code requirements for the specified operating conditions.

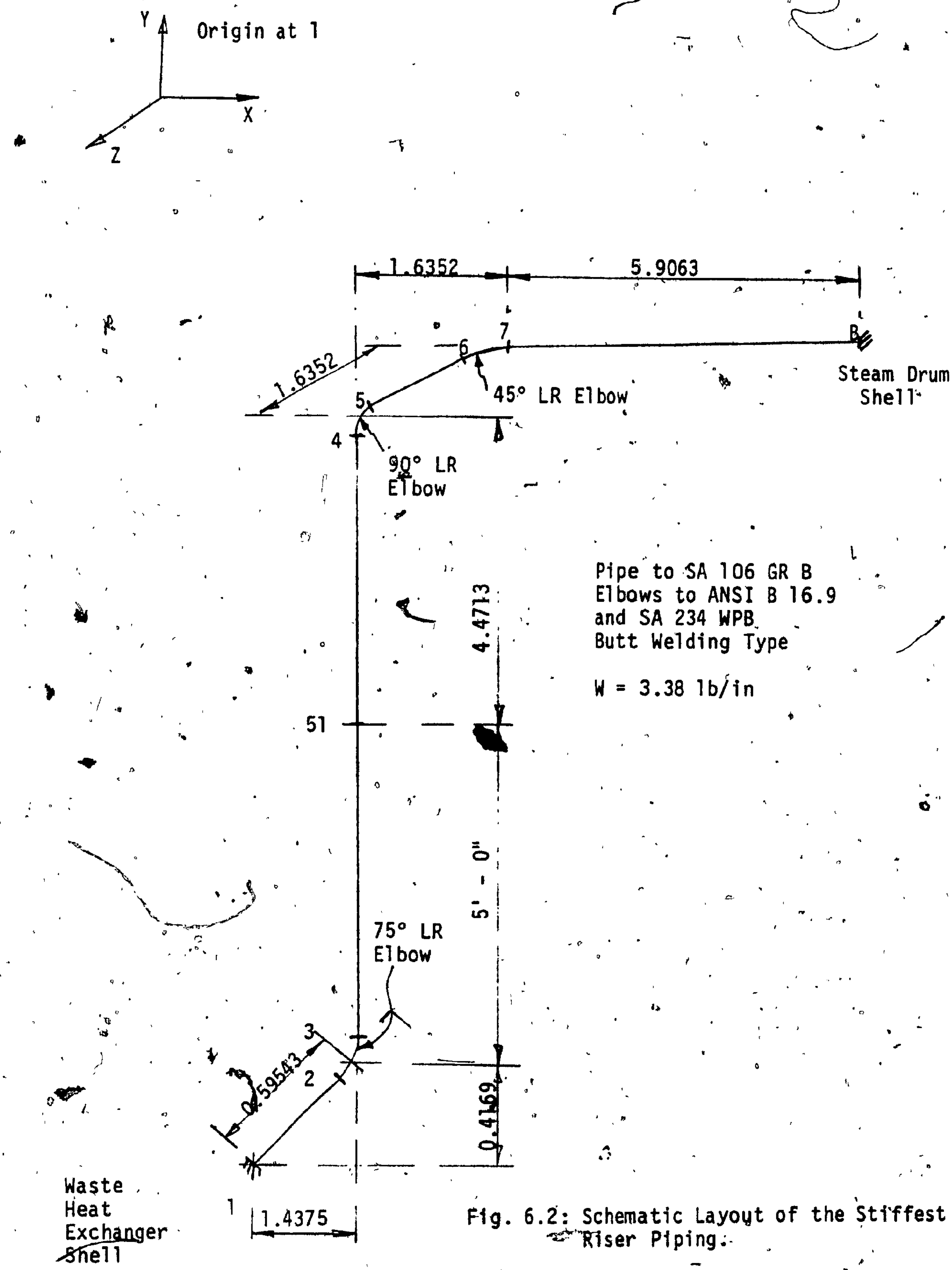


Fig. 6.2: Schematic Layout of the Stiffest Riser Piping.

REFERENCES

REFERENCES

1. Roderick, D.J., Murray, M.V., and Wall, A.G., "Heat Transfer and Draught Loss in the Tube Banks of Shell Boilers", Journal of the Institute of Fuel, October 1959.

2. Kern, D.Q., Process Heat Transfer, McGraw-Hill, 1950.

3. Eckert, E.R.G., Drake, Jr., R.M., Analysis of Heat and Mass Transfer, McGraw-Hill, 1972.

4. Babcock and Wilcox Ltd., Steam: Its Generation and Use.

5. Smith, J.O., McCarthy, J.H., Boelter, W.A., and Brennan, J.H., "Sodium Flow Induced Vibration in Steam Generator Tubes", ASME Paper 64-WA/PWR-4, 1964.

6. Coit, R.L., Peake, C.C., and Lohmeier, A., "Design and Manufacture of Large Surface Condensers - Problems and Solutions", Proc. AM. Power Conference 28, 1966.

7. Proceeding of the Sodium Components Information Meeting, Held in Palo Alto, California, August 20-21, 1963, SAN-8002 Reactor Technology, TID-4500, 34th Ed.

8. Easterling, K.E., "Investigating Failures in Feed Heaters", Engrg. 194, 1964.

9. Reavis, J.R., "WVI-Westinghouse Vibration Correlation for Maximum Fuel Element Displacement in Parallel Turbulent Flow", AM. NUC. SOC. Paper, 13th Annual Meeting, held in San Diego, California, 1967.

10. Paidoussis, M.P., "Vibration of Flexible Cylinders with Supported Ends, Induced by Axial Flow", Proc. Instn. Mech.

Engrg., 180, 1965-1966.

11. Standards of Tubular Exchanger Manufacturers Association, TEMA, Sixth Edition, 1978.

12. Connors, Jr., H.J., Fluidelastic Vibration of Tube Arrays Excited by Cross Flow, Winter Annual Meeting of ASME, December 1, 1970.

13. ASME Boiler and Pressure Vessel Code, Section VIII, Division 1, 1977.

14. Refinery Piping Code ANSI B 31.3, 1976.

15. Blodgett, O.W., "Design of Welded Structures", 1966.

16. Taylor Forge, "Modern Flange Design", Bulletin 502.

17. Barp, B., Angehrn, R., "Calculation of Tube Plates in Heat Exchangers".

18. DeSalvo, G.J., and Swanson, J.A., "Ansys Engineering Analysis System User's Manual", Swanson Analysis System Inc., 1975.

19. Roark, R.J., and Young, W.C., "Formulae for Stress and Strain", McGraw-Hill Inc., 1975.

20. Pettigrew, M.J., Sylvestre, Y., Campagna, A.O., "Vibration Analysis of Heat Exchanger and Steam Generator Designs", AECL 6106.

21. Owens, P.R., "Buffeting Excitation of Boiler Tube Vibration", J. Mechanical Engineering Science, Vol. 7, No. 4, 1965.

22. Thorngrew, J.T., "Predict Exchanger Tube Vibration", Hydrocarbon Processing, April 1970.

23. U.S. Department of the Interior, Bureau of Reclamation, Engineering Monograph No. 27, "Moments and Reactions for Rectangular Plates".

24. Wichman, K.R., Hopper, A.G., and Mershon, J.L., "Local Stresses in Spherical and Cylindrical Shells due to External Loading", Welding Research Council Bulletin, No. 107, August 1965.

25. Adlpipe A Computer Code Developed and Owned by Arthur D. Little Inc., U.S.A.

26. Bijlaard, P.P., and Cranch, E.T., "Interpretive Commentary on the Application of Theory to Experimental Results for Stresses and Deflections due to Local Loads on Cylindrical Shells", Welding Research Council Bulletin, No. 60, 1-2, May 1965.

27. Dally, J.W., "An Experimental Investigation of the Stresses Produced in Spherical Vessels by External Loads Transferred by a Nozzle", ibid., No. 84, January 1963.

28. Roark, R.J., Formula for Stresses and Strain, McGraw-Hill Inc.

APPENDIX A

APPENDIX A

DESIGN OF GAS INLET AND OUTLET CONES

A.1 GAS INLET CONE

The gas inlet cone is shown on drawing No. 8, Appendix C. The cones were designed to ASME Section VIII, Division 1.

A.1.1 Subshell Thickness

As per Paragraph UG 27(c)

$$t = \frac{PR}{SE - 0.6P} + C$$

$P = 40$ psig

$R = 50.9375$ in

$S = 17\ 500$ psi For SA 516 GR 70 Plate

$E = 1$

$C = 0.125$ in

$$t = \frac{40 \times 50.9375}{17\ 500 - 0.6 \times 40} + 0.125$$

$$= 0.1166 + 0.125 = 0.242 \text{ in}$$

Use $^5/_8$ in thick plate since flange calculations govern - see later.

A.1.2 Cone Thickness

Using Paragraph UG 32(g)

$$t = \frac{PD}{2 \cos \alpha \ (SE - 0.6P)} + C$$

$D = 101.875$

$\alpha = 10.6°$

$$t = \frac{40 \times 101.875}{2 \cos 10.6° \ (17\ 500 - 0.6 \times 40)} + 0.125$$

$$= 0.1186 + 0.125$$

$$= 0.244 \text{ in}$$

To include all the reinforcement in cone thickness $t_{REQD} = 244 + 0.1186 =$ 0.363 in.

i.e. Double the reqired thickness.

Hence use of ½ in thk plate from stock material is acceptable.

A.1.3 Gas Inlet Cone Welding End

This is cylindrical section

R = 26.625 in

$$t = \frac{PR}{SE - 0.6P} + C$$

$$= \frac{40 \times 26.625}{17\ 500 - 0.6 \times 40} + 0.125$$

$$= 0.061 + 0.125$$

$$= 0.186 \text{ in}$$

Again, use of ½ in thk plate is acceptable.

A.1.4 Cone Reinforcement

We will now check if additional reinforcement is required at cone to shell sections using ASME Section VIII, Division 1, Appendix 1, Paragraph UA 5, UA 5(b), Large End.

$$\frac{P}{SE} = \frac{40}{17\ 500 \times 1} = 0.0023$$

$$\Delta = 15.9° \text{ from Table UA 5-1}$$

$\Delta > \alpha = 10.5°$. Hence, no additional reinforcement is required at cone to cylinder junction at the large end.

UA 5(c). small end

$$\frac{P}{SE} = \frac{40}{17\ 500 \times 1} = 0.0023$$

From Table UA 5-2 $\Delta = 4.2°$ $< \alpha = 10.5°$.

Hence, we must check for reinforcement

$$A_{REQD} = \frac{PR_s^2\ K}{2\ SE} \left(1 - \frac{\Delta}{\alpha}\right) \tan\ \alpha$$

$$= \frac{40 \times 26.625^2}{2 \times 17\ 500 \times 1} \times 1 \left(1 - \frac{4.2}{10.5}\right) \tan 10.5$$

$$= 0.09\ in^2$$

Area available $= A_e = M \sqrt{R_s t} \left[\left(t_c - \frac{t}{\cos\alpha}\right) + \left(t_s - t\right)\right]$

Where M = smaller of $\left[\frac{t_s}{t} \cos(\alpha - \Delta)\right]$ or $\left[\frac{t_e \cos\alpha \cos(\alpha - \Delta)}{t}\right]$

M = smaller of $\left[\frac{0.375}{0.1186} \cos\ 6.3\right]$ or $\left[\frac{0.375 \cos 10.5 \cos 6.3}{0.1186}\right]$

= smaller of 3.143 or 3.09

M = 3.09

$$\therefore A_e = 3.09 \sqrt{26.625 \times 0.1186} \left[\left(0.375 - \frac{0.1186}{\cos 10.5}\right) + (0.375 - 0.1186)\right]$$

$$= 2.80\ in^2 > A_{REQD}$$

A.1.5 Flange Design

Design of gas inlet cone main flange for P = 40 psig is shown in Table A.1.

To check the main flange for operating pressure and external moment the procedure outlined in section 4.2.4(b), Case 2 is used.

Design of Gas Inlet and Outlet Cone Flanges for Design Pressure of 40 psig.

TABLE A.1

WELDING NECK FLANGE DESIGN — SHEET A

DESIGN CONDITIONS		GASKET and BOLTING CALCULATIONS		FROM Fig UA 49.1 and UA 49.2
Design Pressure, P	40 psig	Gasket Details: 104.125" I/D x 105.125" O/D x 3/16 thk SS 410 Flat Metal Jckd Asbestos	Facing Details: 105.75" O/D x $\frac{1}{8}$" Raised Face	N = 0.5
Design Temperature	650°F			b = 0.25
Flange Material	SA 516 GR 65			y = 9000
Bolting Material	SA 193 GR B7			m = 3.75

Corrosion Allowance	0.125	$W_{m2} = b\pi G y$ = 739 551	A_m = greater of W_{m1}/S_b or W_{m2}/S_a = 29.58
Allowable Stresses: Flange	Design Temp, S_{fo} = 16 300	$H_p = 2b\pi G m P$ = 24 652	A_b = 72 x 0.419 = 30.17 in^2
	Atm Temp, S_{fa} = 16 300	$H = G^2\pi P/4$ = 343 891	$W = \frac{1}{2}(A_m + A_b)S_a$ = 746 850
Bolting	Design Temp, S_b = 25 000	$W_{m1} = H_p + H$ = 368 543	W_{m1} =
	Atm Temp, S_a = 25 000	Gasket Width Check (Raised Face ONLY) $N_{min} = A_b S_a / 2y\pi G$ = 0.127	

CONDITION	LOAD	x LEVER ARM	= MOMENT
Operating	$H_D = \pi B^2 P/4$ = 327 653	$h_D = R + .5g_1$ = 2.1875	$M_D = H_D h_D$ = 716 741
	$H_G = W_{m1} - H$ = 24 562	$h_G = .5(C - G)$ = 1.125	$M_G = H_G h_G$ = 27 632
	$H_T = H - H_D$ = 16 238	$h_T = .5(R + g_1 + h_G)$ = 1.8125	$M_T = H_T h_T$ = 29 431
			M_o = 773 804
Gasket Seating	$H_G = W$ = 746 850	$h_G = .5(C - G)$ = 1.125	M_o = 840 206

Allowable Stress	STRESS CALCULATION — Operating Conditions (use M_o)		
1.5 S_{fo}	Long Hub, $S_H = fM_o/\lambda g_1^2$	20 524	
S_{fo}	Radial Flg, $S_R = \beta M_o/\lambda t^2$	1134	
S_{fo}	Tang Flg, $S_T = (M_o Y/t^2) - ZS_R$	3290	
S_{fo}	greater of $.5(S_H + S_R)$ or $.5(S_H + S_T)$	11 907	

Allowable Stress	STRESS CALCULATION — Gasket Seating (use M_o)		
1.5 S_{fa}	Long Hub, $S_H = fM/\lambda g_1^2$	22 2[illegible]	
S_{fa}	Radial Flg, $S_R = \beta M/\lambda t^2$	12[illegible]	
S_{fa}	Tang Flg, $S_T = (MY/t^2) - ZS_R$	35[illegible]	
S_{fa}	greater of $.5(S_H + S_R)$ or $.5(S_H + S_T)$	12 9[illegible]	

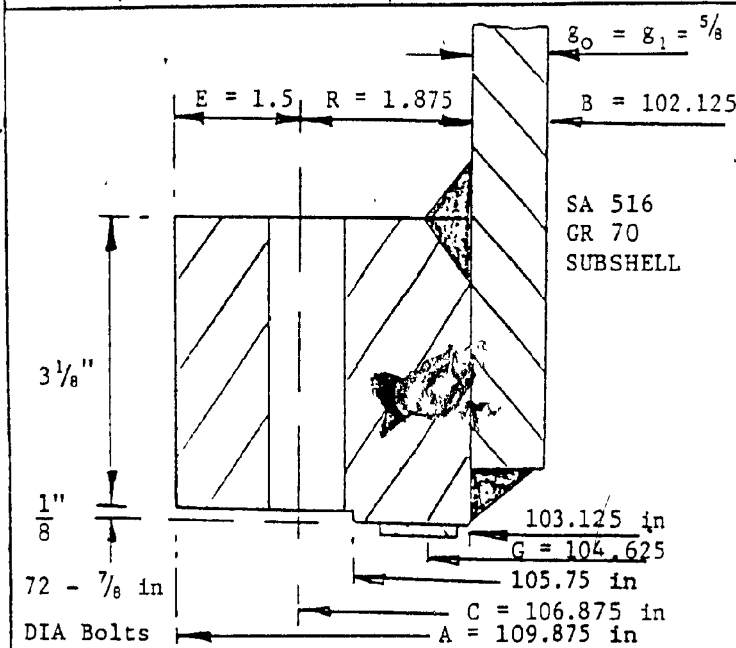
SHAPE CONSTANTS (From design table 2 and design charts 1, 2 & 5)

K = A/B = 1.0786	h/h_o =
T = 1.88	F = 0.9089
Z = 13.23	V = 0.5511
Y = 25.5	f = 1
U = 28.03	$e = F/h_o$ = 0.1139
g_1/g_o = 1	$d = \frac{U}{V} h_o g_o^2$ = 158.863
$h_o = \sqrt{Bg_o}$ = 7.979	

OTHER STRESS FORMULA FACTORS

t (assumed)	3.25
$\alpha = te + 1$	1.3702
$\beta = 4/3\, te + 1$	1.494
$\gamma = \alpha/T$	0.729
$\delta = t^3/d$	0.2161
$\lambda = \gamma + \delta$	0.9451
$M = M_o/B$	7577
$M = M_o/B$	8227

If bolt spacing exceeds 2a + t, multiply M_o and M_a in above equations by $\sqrt{\frac{\text{Bolt spacing}}{2a + t}}$



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Computed JSK Date Feb 14/79

Checked Number

Design of Gas Inlet Cone Flange for Operating Pressure and External Moment i.e. For Equivalent Pressure of 60 PSIG

TABLE A.2

WELDING NECK FLANGE DESIGN SHEET A

DESIGN CONDITIONS		GASKET and BOLTING CALCULATIONS		FROM Fig UA 49.1 and UA 49.2
Design Pressure, P	60 psig	Gasket Details: 104.125" I/D x 105.125" O/D x 3/16 thk SS 410 flat Metal Jckd Asbestor	Facing Details: 105.75" O/D x 1/8" Raised Face	N = 0.5
Design Temperature	650°F			b = 0.25
Flange Material	SA 516 GR 65			y = 9000
Bolting Material	SA 193 GR B7			m = 3.75

Corrosion Allowance		0.125		
Flange	Design Temp., S_{fo}	16 300	$W_{m2} = b\pi G y$ = 739 551	A_m = greater of W_{m2}/S_a or W_{m1}/S_b = 29.58 in^2
	Atm. Temp., S_{fa}	16 300	$H_p = 2b\pi GmP$ = 36 977	A_b = 72 x 0.419 = 30.17 in^2
Bolting	Design Temp., S_b	25 000	$H = G^2\pi P/4$ = 515 836	$W = S(A_m + A_b)S_a$ = 746 850
	Atm. Temp., S_a	25 000	$W_{m1} = H_p + H$ = 552 814	W_{m1} =
			Gasket Width Check (Raised Face ONLY) $N_{min} = A_b S_a / 2y\pi G$ = 0.127	

CONDITION	LOAD	x LEVER ARM	= MOMENT
Operating	$H_D = \pi B^2 P/4$ = 491 479	$h_D = R + .5g_1$ = 2.1875	$M_D = H_D h_D$ = 1075 110
	$H_G = W_{m1} - H$ = 36 977	$h_G = .5(C - G)$ = 1.125	$M_G = H_G h_G$ = 41 599
	$H_T = H - H_D$ = 24 357	$h_T = .5(R + g_1 + h_G)$ = 1.8125	$M_T = H_T h_T$ = 44 147
			M_o = 1160 856
Gasket Seating	$H_G = W$ = 746 850	$h_G = .5(C - G)$ = 1.125	M_a = 840 206

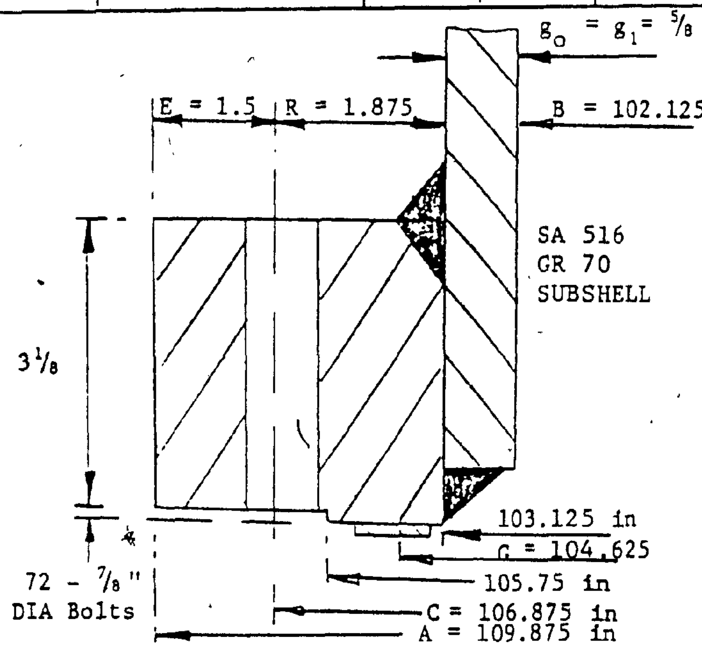
Allowable Stress	STRESS CALCULATION — Operating (use M_o)			
1.5 S_{fo}	Long. Hub, $S_H = fM_o/\lambda g_1^2$	30 787		
S_{fo}	Radial Flg., $S_R = \beta M_o/\lambda t^2$	1139		
S_{fo}	Tang. Flg., $S_T = (M_o Y/t^2) - ZS_R$	12 377		
S_{fo}	greater of .5($S_H + S_R$) or .5($S_H + S_T$)	21 583		

Allowable Stress	STRESS CALCULATION — Gasket Seating (use M)			
1.5 S_{fa}	Long. Hub, $S_H = fM/\lambda g_1^2$	22 284		
S_{fa}	Radial Flg., $S_R = \beta M/\lambda t^2$	1231		
S_{fa}	Tang. Flg., $S_T = (MY/t^2) - ZS_R$	3572		
S_{fa}	greater of .5($S_H + S_R$) or .5($S_H + S_T$)	12 928		

SHAPE CONSTANTS (From design table 2 and design charts 1, 2 & 5)			
K = A/B	1.0786	h/h_o =	
T	1.88	F	0.9089
Z	13.23	V	0.5511
Y	25.5	f	1
U	28.03	$e = F/h_o$	0.1139
g_1/g_o	1	$d = \frac{U}{V} h_o g_o^2$	158.863
$h_o = \sqrt{Bg_o}$	7.979		

OTHER STRESS FORMULA FACTORS			
t (assumed)	3.25		
$\alpha = te + 1$	1.3702		
$\beta = 4/3\ te + 1$	1.494		
$\gamma = \alpha/T$	0.729		
$\delta = t^3/d$	0.2161		
$\lambda = \gamma + \delta$	0.9451		
$M_o = M_o/B$	11367		
$M = M_a/B$	8227		

If bolt spacing exceeds 2a + t, multiply M_o and M_a in above equations by $\sqrt{\frac{\text{Bolt spacing}}{2a + t}}$



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Computed JSK Date

Checked Number

Thus the joint is checked for a total equivalent pressure of 60 psi which includes operating pressure and specified external loadings including earthquake as per Table 4.2.

$$S_H = \frac{fM_o}{\lambda g_1^2 B} + \frac{PB}{4g_o}$$

$$= \frac{1 \times 1\,160\,856}{0.9451 \times 0.625^2 \times 102.125} + \frac{15.3 \times 102.125}{4 \times 0.625}$$

$$= 30\,789 + 625$$

$$= 31\,415 \text{ psi}$$

This solution is shown in Table A.2. Definition of above symbols are also given in Table A.2. Allowable for S_H = 1.5 S for pressure, thermal and deadweight loading.

S_H = 1.33 x 1.5 S for pressure, thermal, deadweight and earthquake loading.

= 1.33 x 1.5 x 17 500 = 34 912 psi

S_R = 1139 psi

Allowable for S_R = 1.33 x 1.5 S = 1.33 x 1.5 x 16 300
= 32 600 psi

S_T = 12 377 psi

Allowable for S_T = 1.33 x 1.5 S = 1.33 x 1.5 x 16 300
= 32 600 psi

A.1.6 Alternative Assessment of Gas Inlet Cone Flange

In accordance with Paragraph UA 48 (3) of the code, the cone flange may be classified as an optional type flange since

g_o = 0.625 is not greater than 5/8 in

$$\frac{B}{g_o} = \frac{102.125}{0.625} = 163.4 \text{ is not greater than } 300$$

P_{max} = 60 psi is not greater than 300 psi

Operating temperature = 375° which is not greater than 700°F.

Using Paragraph UA 51(b) for an optional type flange

S_R = radial stress = 0 $\quad$ S_H = longitudinal hub stress = 0

S_T = calculated tangential stress in flange

$$S_T = \frac{M_O Y}{B t_F^2}$$

$$\therefore \text{ maximum } S_T = \frac{1\ 160\ 856 \times 25.5}{102.125 \times 3.25^2} = 27\ 442 \text{ psi}$$

Again the allowable stress used is as per ASME Section III, Class 2.

S_A = 1.5 S = 1.5 x 16 300 = 24 450 psi for pressure, thermal and deadweight loading.

Actual stress for this load case is

$$27\ 442 \times \frac{51}{60} = 23\ 325 \text{ psi}$$

S_A = 1.33 x 1.5 S = 1.33 x 1.5 x 16 300

= 32 000 psi

A.1.7 Other Flanges

In all other cases on the gas inlet cone the flanges used were to ANSI B 16.5 and SA 105 for pressure rating of 150 psi.

A.2 GAS OUTLET CONE

The gas outlet cone is shown on drawing No. 2, in Appendix C.

A.2.1 Subshell Thickness

Calculation procedure is identical to Section A.1.1

$t_{REQUIRED} = 0.242$ in

$t_{SUPPLIED} = \frac{5}{8}$ in.

A.2.2 Cone Thickness

As Paragraph UG 32(g)

$$t = \frac{PD}{2 \cos \alpha \, (SE - 0.6P)} + C$$

$P = 40$ psi

$D = 101.875$ in

$\alpha = 15°$

$S = 17\,500$ psi For SA 516 GR.70

$E = 1$

$C = 0.125$ in

$$t = \frac{40 \times 101.875}{2 \cos 15 \, (17\,500 - 0.6 \times 40)} + 0.125$$

$$= 0.121 + 0.125$$

$$= 0.246 \text{ in}$$

To include all the required reinforcement in shell

$t_{REQD} = 0.246 + 0.121$

$= 0.367$ in

Hence, $t = \frac{1}{2}$ in thk is acceptable.

A.2.3 Gas Outlet Cone Welding End

Using Paragraph UG 27(c) for cylindrical section

$$t = \frac{PR}{SE - 0.6P} + C$$

$$R = 17.625 \text{ in}$$

$$t = \frac{40 \times 17.625}{17\ 500 - 0.6 \times 40} + 0.125$$

$$= 0.04034 + 0.125$$

$$= 0.1653 \text{ in}$$

$$t_{ACTUAL} = \frac{1}{2} \text{ in thk}$$

A.2.4 Cone Reinforcement

Again, we will check if additional reinforcement is required at cone to shell sections using ASME Section VIII, Division 1, Appendix 1, Paragraph UA 5.

UA 5(b) Large End

$$\frac{P}{SE} = \frac{40}{17\ 500 \times 1} = 0.0023$$

Δ = 15.9° from Table UA 5-1

$\Delta > \alpha = 15°$ Hence no additional reinforcement is required at cone to cylinder junction at the large end.

UA 5(c) Small End.

$$\frac{P}{SE} = \frac{40}{17\ 500 \times 1} = 0.0023$$

From Table UA 5-2 $\Delta = 4.2°$

$$\Delta = 4.2° < 15°$$

Thus we must check for reinforcement

$$A_{REQUIRED} = \frac{PR_s^2 K}{2\,SE}\left(1 - \frac{\Delta}{\alpha}\right)\tan\alpha$$

$$= \frac{40 \times 17.625^2 \times 1}{2 \times 17\,500 \times 1}\left(1 - \frac{4.2}{15}\right)\tan 15$$

$$= 0.0685 \text{ in}^2$$

$$\text{Area available} = A_e = M\sqrt{R_s t}\left[\left(t_c - \frac{t}{\cos\alpha}\right) + \left(t_s - t\right)\right]$$

$$\text{Where } M = \text{smaller of } \left[\frac{t_s}{t}\cos(\alpha - \Delta)\right] \text{ or } \left[\frac{t_e \cos\alpha\cos(\alpha-\Delta)}{t}\right]$$

$$= \text{smaller of } \left[\frac{0.375}{0.121}\cos 10.8\right] \text{ or } \left[\frac{0.375\cos 15\cos 10.8}{0.121}\right]$$

$$= \text{smaller of } 3.044 \text{ or } 2.94$$

$$M = 2.94$$

$$\therefore A_e = 2.94\sqrt{17.625 \times 0.121}\left[\left(0.375 - \frac{0.121}{\cos 15}\right) + (0.375 - 0.12)\right]$$

$$= 2.163 \text{ in}^2$$

$$> A_{REQUIRED}$$

A.2.5 Flange Design

The main flange on gas outlet cone was calculated for a design pressure of 40 psig and gasket seating using an identical procedure to that shown in A.1.5, page   The gas outlet cone flange is not subjected to any significant external moments since expansion joints are provided on the gas outlet line to the electro static precipitators. Thus the gas outlet cone flange was not required to be checked for such external moments and forces.

Once again, all other flanges for manways and instrument connections were in accordance with ANSI B 16.5 and SA 105 for a pressure rating of 150 psig.

APPENDIX B

APPENDIX B

DESIGN OF STEAM DRUM

Details of the steam drum are shown on drawing No. 3 in Appendix C. The steam drum pressure parts were designed in accordance with ASME Section VIII, Division 1.

B.1 SHELL THICKNESS

As per Paragraph UG 27(c)

$$t = \frac{PR}{SE - 0.6P} + C$$

$P = 180$ psi

$R = 24$ in

$S = 17\ 500$ psi For SA 516 GR 70

$E = 1$

$C = 0.125$ in

$$t = \frac{180 \times 24}{17\ 500 - 0.6 \times 180} + 0.125$$

$$= 0.2484 + 0.125$$

$$= 0.373 \text{ in}$$

The additional thickness needed to include all the required reinforcement for openings in shell is 0.2483 in

$$t_{REQD} = 0.373 + 0.2483$$

$$= 0.6217 \text{ in}$$

Use $T = \frac{5}{8}$ in thk

The distance between nozzle centres will be not less than sum of inside diameters.

B.2 PLAIN HEAD THICKNESS

Using Paragraph UG 32(d)

$$t = \frac{PD}{2SE - 0.2P} + C \qquad \text{For 2:1 Semi Elliptical Head}$$

D = inside diameter of head

$$t = \frac{180 \times 48}{2 \times 17\,500 - 0.2 \times 180} + 0.125$$

$$= 0.247 + 0.125$$

$$= 0.372 \text{ in}$$

This head will not have any connections larger than 2 in diameter

$T = \frac{3}{8}$ in minimum thickness was selected

B.3 THICKNESS OF FLUED MANHOLE

This head was calculated in accordance with ASME Section 1, Paragraph Pg 29.1 and Pg 29.3.

$$t = \frac{5\ PL}{4.8\ SE} + C$$

Dishing Radius, $L = 0.8D = 0.8 \times 48$

$$t = \frac{5 \times 180 \times 0.8 \times 48}{4.8 \times 17\,500} + 0.125$$

$$= 0.4114 + 0.125$$

Allow 15% or minimum of $\frac{1}{8}$ in for opening

$$t_{REQD} = 0.4114 + 0.125 + 0.125$$

$$= 0.6614 \text{ in}$$

Use $T = \frac{11}{16}$ in minimum thickness for flued head

B.4 TYPICAL NOZZLE NECK THICKNESS CALCULATION

Using Paragraph UG 27(c)

$$t = \frac{PR}{SE - 0.6P} + C$$

Consider 12 in nozzle

$P = 180$ psi

$R = 5.875$ in

$S = 15\ 000$ psi For SA 106 GR B Pipe

$C = 0.125$

$E = 1$

$$t = \frac{180 \times 5.875}{15\ 000 - 0.6 \times 180} + 0.125$$

$$= 0.0710 + 0.125$$

$$= 0.196 \text{ in}$$

Allow 12½% manufacturing margin on pipe

$$t = 0.196 \times 1.125$$

$$t_{NOM} = 0.2205 \text{ in}$$

Use 12" x-STG Pipe

$$t_{NOM} = 0.50 \text{ in}$$

$$t_{MIN} = 0.4375 \text{ in}$$

B.5 CALCULATION OF ALLOWABLE LOADS FOR STEAM OUTLET NOZZLE ON DRUM

The local stresses at all nozzles subjected to external loads and moments were investigated using Welding Research Council Bulletin, No. 107 entitled "Local Stresses in Speherical and Cylindrical Shells due to External Loadings" [24]. The approach presented in this paper is based on analytical work accomplished by Prof. P.P. Bijlaard of Cornell University. This theoretical work has been further verified by experimental work reported in references [26] and [27].

As an application of WRC Bulletin No. 107, the calculation of allowable loads for 12 in steam outlet nozzle on the steam drum is presented as follows:

Attachment Parameter, $\beta = \dfrac{0.875\ r_o}{R_M}$

r_o = outside radius of cylindrical attachment = 6.375 in

R_M = mean radius of cylindrical shell = 24.375 in

$\therefore\ \beta = \dfrac{0.875 \times 6.375}{24.375} = 0.229$

Shell Parameter, $\gamma = \dfrac{R_M}{T}$

T = wall thickness of cylindrical shell

= 0.5 in

R_M = 24.375 in

$\gamma = \dfrac{24.375}{0.5} = 48.75$

The local stress in the shell calculated using arbitary loads are shown in Table A.3. The stresses in the shell due to internal pressure are

Hoop Stress, $\sigma_\phi = \dfrac{PR_M}{T}$

$= \dfrac{0.155 \times 24.375}{0.5}$

= 7.62 KSI

Longitudinal Stress, $\sigma_L = \frac{PR_M}{2T}$

$= 3.82$ KSI

Then using definitions in Table A.3, we have by inspection:

1. Maximum allowable primary loads

$P_c = 2.5$ KIP

$M_c = 40$ in KIP

$M_L = 50$ in KIP

$M_T = 50$ in KIP

$V_c = 2.5$ KIP

$V_L = 2.5$ KIP

Which will produce a maximum hoop stress of $\sigma_\phi = 7.63 + 17.5$

$= 25.11$ KSI

2. Maximum allowable range of primary plus secondary loads are:

$P = 5$ KIP

$M_c = 75$ in KIP

$M_L = 100$ in KIP

$M_T = 100$ in KIP

$V_c = 5$ KIP

$V_L = 5$ KIP

Which will give maximum range of hoop stress of $\sigma_\phi = 7.63 + 33.62$

$= 40.95$ KSI

Allowable stress range $= 3\ S_M$

$= 3 \times 21.7$

$= 65.1$ KSI

Table 5—Computation Sheet for Local Stresses in Cylindrical Shells

1. Applied Loads[b]

Radial load,	P	5 ~~lb.~~ K
Circ. Moment,	Mc	75 in. ~~lb.~~ K
Long. Moments,	ML	100 in. ~~lb.~~ K
Torsion Moment,	MT	100 in. ~~lb.~~ K
Shear Load,	Vc	2 ~~lb.~~ K
Shear Load,	VL	5 ~~lb.~~ K

2. Geometry

Vessel thickness,	T	0.5 in.
Attachment radius,	r_o	6.375 in.
Vessel radius,	R_m	24.375 in.

3. Geometric Parameters

$\gamma = \frac{R_m}{T} = 48.75$

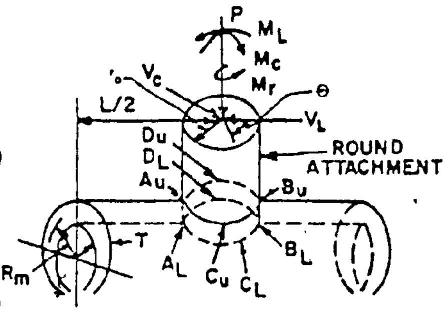
$\beta = (0.875)\frac{r_o}{R_m} = 0.229$

Stress Concentration due to.

a) membrane load, Kn ——

b) bending load, Kb ——

*NOTE. Enter all force values in accordance with sign convention



CYLINDRICAL SHELL

From Fig.	Read curves for	Compute absolute values of stress and enter result ·	Au	AL	Bu	BL	Cu	CL	Du	DL
			STRESSES – if load is opposite that shown, reverse signs shown							
3C	$\frac{N\phi}{P/R_m}$ 3.5	$K_n\left(\frac{N\phi}{P/R_m}\right)\cdot\frac{P}{R_mT}$ 1.436	-1.44	-1.44	-1.44	-1.44	-1.44	-1.44	-1.44	-1.44
1C	$\frac{M\phi}{P}$ 0.06	$K_b\left(\frac{M\phi}{P}\right)\cdot\frac{6P}{T^2}$ = 7.2	-7.2	+7.2	-7.2	+7.2	-7.2	+7.2	-7.2	+7.2
3A	$\frac{N\phi}{M_c/R_m^2\beta}$ 1.9	$K_n\left(\frac{N\phi}{M_c/R_m^2\beta}\right)\cdot\frac{M_c}{R_m^2\beta T}$ 2.1					-2.1	-2.1	+2.1	+ 2.1
1A	$\frac{M\phi}{M_c/R_m\beta}$ 0.07	$K_b\left(\frac{M\phi}{M_c/R_m\beta}\right)\cdot\frac{6M_c}{R_m\beta T^2}$ 22.58					-22.58	+22.58	+22.58	22.58
3B	$\frac{N\phi}{M_L/R_m^2\beta}$ 4.3	$K_n\left(\frac{N\phi}{M_L/R_m^2\beta}\right)\cdot\frac{M_L}{R_m^2\beta T}$ 6.32	-6.32	-6.32	+6.32	+6.32				
1B or 1B-1	$\frac{M\phi}{M_L/R_m\beta}$ 0.024	$K_b\left(\frac{M\phi}{M_L/R_m\beta}\right)\cdot\frac{6M_L}{R_m\beta T^2}$ 10.32	-10.32	+10.32	+10.32	-10.32				
Add algebraically for summation of ϕ stresses, σ_ϕ =			+25.28	+9.76	8.0	1.76	-33.32	+21.14	16.04	14.72
4C	$\frac{N_x}{P/R_m}$ 6.1	$K_n\left(\frac{N_x}{P/R_m}\right)\cdot\frac{P}{R_mT}$ = 2.5	-2.5	-2.5	-2.5	-2.5	-2.5	-2.5	-2.5	- 2.5
2C	$\frac{M_x}{P}$ 0.036	$K_b\left(\frac{M_x}{P}\right)\cdot\frac{6P}{T^2}$ = 4.32	-4.32	+4.32	-4.32	+4.32	-4.32	+4.32	-4.32	+4.32
4A	$\frac{N_x}{M_c/R_m^2\beta}$ 4.3	$K_n\left(\frac{N_x}{M_c/R_m^2\beta}\right)\cdot\frac{M_c}{R_m^2\beta T}$ = 4.74					-4.74	-4.74	+4.74	+4.74
2A	$\frac{M_x}{M_c/R_m\beta}$ 0.03	$K_b\left(\frac{M_x}{M_c/R_m\beta}\right)\cdot\frac{6M_c}{R_m\beta T^2}$ = 9.99					-9.99	+9.99	+9.99	-9.99
4B	$\frac{N_x}{M_L/R_m^2\beta}$ 1.9	$K_n\left(\frac{N_x}{M_L/R_m^2\beta}\right)\cdot\frac{M_L}{R_m^2\beta T}$ = 2.8	-2.8	-2.8	+2.8	+2.8				
2B or 2B-1	$\frac{M_x}{M_L/R_m\beta}$ 0.03	$K_b\left(\frac{M_x}{M_L/R_m\beta}\right)\cdot\frac{6M_L}{R_m\beta T^2}$ = 12.9	-12.9	+12.9	+12.9	-12.9				
Add algebraically for summation of X stresses, σ_x			-22.52	+11.92	+8.88	-8.28	-21.55	+7.07	7.91	-3.43
Shear stress due to Torsion, M_T		$\tau_{\phi x} = \tau_{x\phi} = \frac{M_T}{2\pi r_o^2 T}$ 0.78	+0.78	+0.78	+0.78	+0.78	+0.78	+0.78	+0.78	+0.78
Shear stress due to load, Vc		$\tau_{x\phi} = \frac{V_c}{\pi r_o T}$ 0.5	+0.5	+0.5	-0.5	-0.5				
Shear stress due to load, VL		$\tau_{x\phi} = \frac{V_L}{\pi r_o T}$ 0.50					-0.5	-0.5	+0.5	+0.5
Add Algebraically for summation of shear stresses, τ =			1.28	1.28	0.28	0.28	0.28	0.28	1.28	1.28
COMBINED STRESS INTENSITY, S										
1) When σ_ϕ & σ_x have like signs		$S = \frac{1}{2}\left[\sigma_\phi + \sigma_x + \sqrt{(\sigma_\phi - \sigma_x)^2 + 4\tau^2}\right]$								
2) When $\tau = 0$		S = largest of σ_ϕ, σ_x or $\lvert\sigma_\phi - \sigma_x\rvert$								
3) When σ_ϕ & σ_x have unlike signs		$S = \sqrt{(\sigma_\phi - \sigma_x)^2 + 4\tau^2}$								

$N_\phi/(M_L/R_m^2\beta)$ so determined by (C_L) from Table 8 (see para. 4.3).

4.2.2.5.2: When considering bending moment (M_L): $\beta = K_L\sqrt[3]{\beta_1\beta_2^2}$ where K_L is given in Table 8.

4.3 Calculation of Stresses

4.3.1 STRESSES RESULTING FROM RADIAL LOAD P.

4.3.1.1 Circumferential Stresses (σ_ϕ):

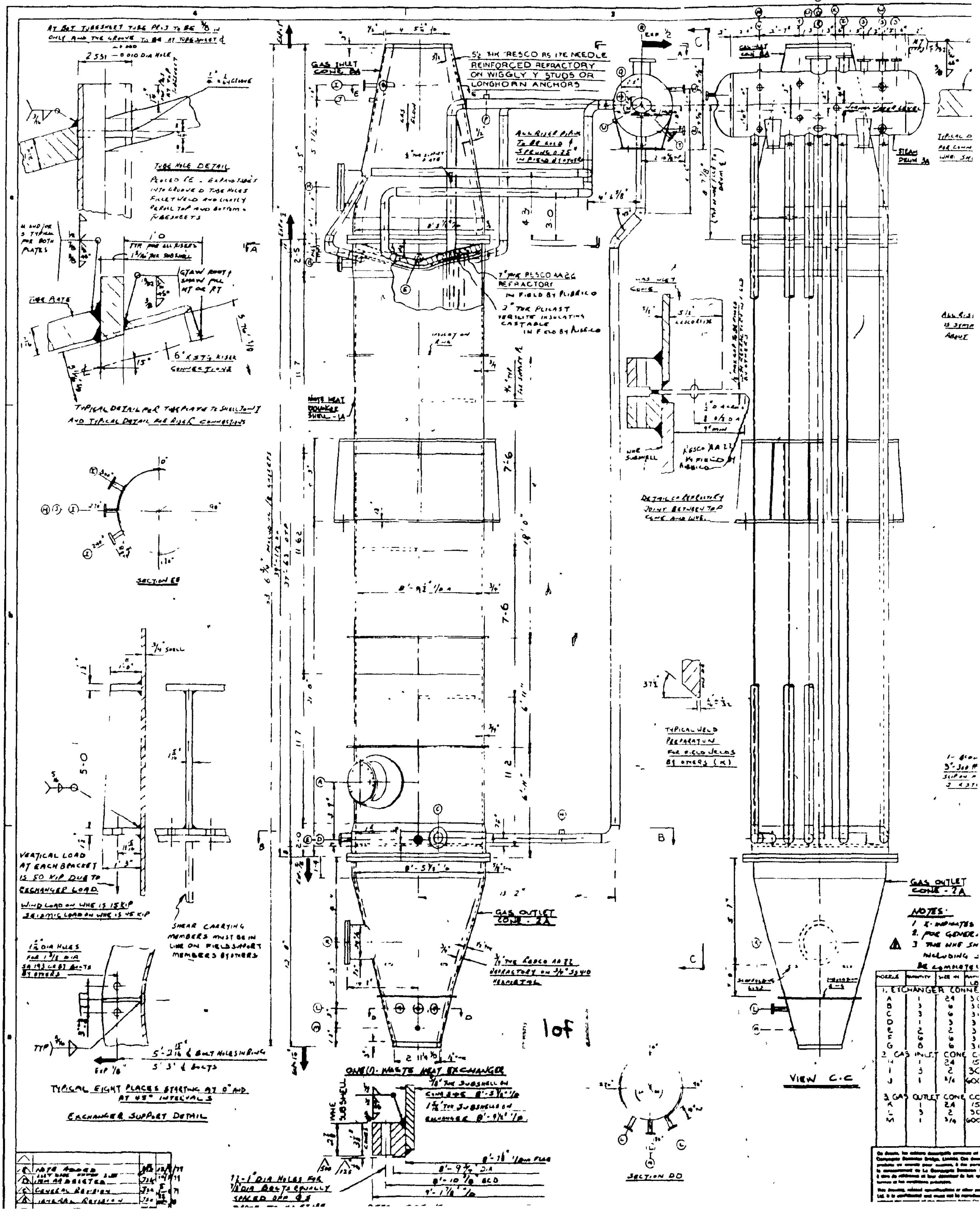
Step 1. Using the applicable values of β and

APPENDIX C

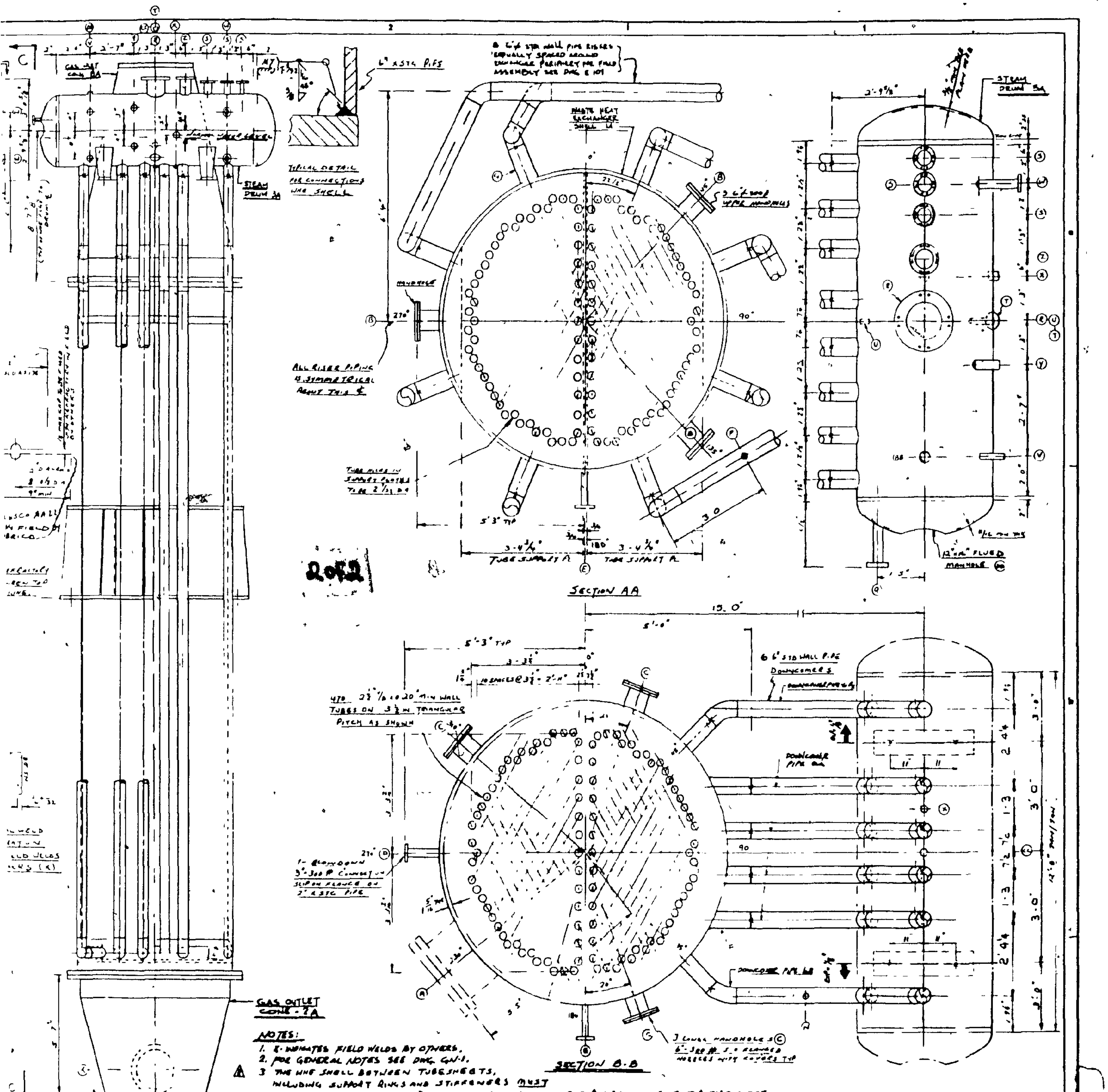
APPENDIX C

The following engineering drawings are included in this Appendix.

DRAWING NUMBER	BRIEF DESCRIPTIONS
GN 1 REV 3	General Notes.
E 1 REV E	General Arrangement.
E 10 REV B	Steam Drum Connections and Internals.
E 101 REV A	Riser and Downcomer Piping Layout.
1 REV 8	WHE Shell and Tubes
2 REV 5	Gas Outlet Cone
5 REV 2	Davits and Baffle Plates
8 REV 5	Gas Inlet Cone
13 REV 1	WHE Lifting Attachments



GAS INLET CONE 8A
5½" THK RESCO RS 17E NEEDLE REINFORCED REFRACTORY ON WIGGLY Y STUDS OR LONGHORN ANCHORS
TUBE HOLE DETAIL
TYPICAL DETAIL FOR TUBEPLATE TO SHELL JOINT AND TYPICAL DETAIL FOR RISER CONNECTIONS
SECTION EE
WASTE HEAT EXCHANGER SHELL - 1A
DETAIL OF REFRACTORY JOINT BETWEEN TOP CONE AND WHE
TYPICAL WELD PREPARATION FOR FIELD WELDS BY OTHERS (K)
VERTICAL LOAD AT EACH BRACKET IS 50 KIP DUE TO EXCHANGER LOAD.
WIND LOAD ON WHE IS 15 KIP
SEISMIC LOAD ON WHE IS 45 KIP
SHEAR CARRYING MEMBERS MUST BE IN LINE ON FIELD SUPPORT MEMBERS BY OTHERS
TYPICAL EIGHT PLACES STARTING AT 0° AND AT 45° INTERVALS
EXCHANGER SUPPORT DETAIL
GAS OUTLET CONE - 2A
ONE(1) - WASTE HEAT EXCHANGER
SECTION DD
VIEW C-C
STEAM DRUM 3A
NOTES
1. EXCHANGER CONNECTIONS
2. GAS INLET CONE
3. GAS OUTLET CONE
1of



NOTES:

1. X - INDICATES FIELD WELDS BY OTHERS.
2. FOR GENERAL NOTES SEE DWG. GN-1.
3. THE WHE SHELL BETWEEN TUBESHEETS, INCLUDING SUPPORT RINGS AND STIFFENERS MUST BE COMPLETELY INSULATED. DO NOT EXTERNALLY INSULATE TOP SUBSHELL ABOVE TOP TUBESHEET.

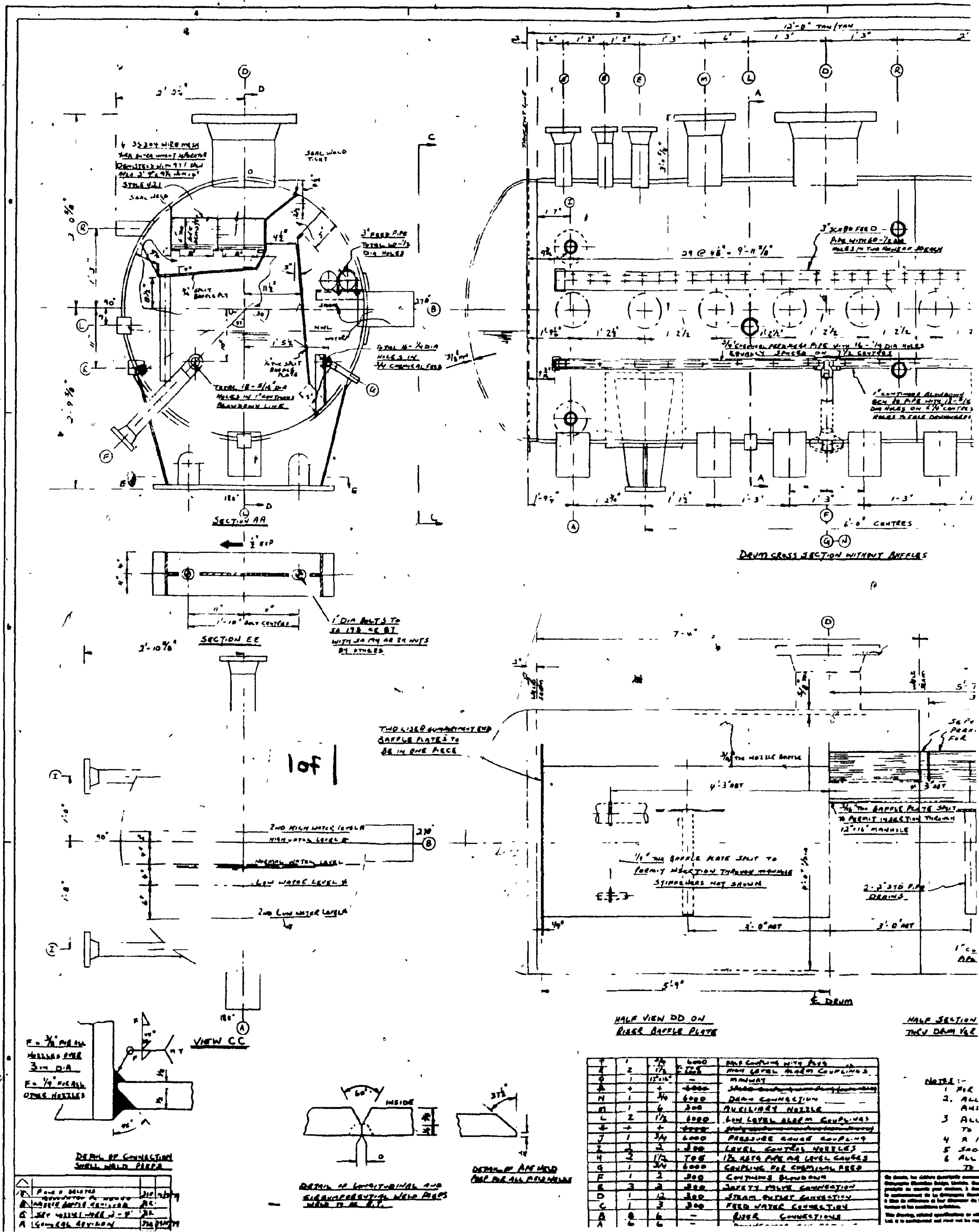
NOZZLE	QUANTITY	SIZE IN	RATING LB	SERVICE	NOZZLE	QUANTITY	SIZE IN	RATING LB	SERVICE
1 EXCHANGER CONNECTIONS					4 PIPING CONNECTIONS				
A	1	24	300	MANWAY	N	1	1	6000	THREADED THERMOWELL CONNECTION ON DOWN COMER
B	3	6	300	TOP TUBE SHEET HAND HOLES					
C	3	6	300	BOTTOM TUBE SHEET HAND HOLES	P	1	1	6000	THREADED THERMOWELL CONNECTION ON RISER
D	1	3	300	BLOWDOWN NOZZLE					
E	2	2	300	CONTROL NOZZLES	5 STEAM DRUM CONNECTIONS				
F	6	6	300	DOWNCOMERS	Q	1	3	300	FEED WATER CONNECTION
G	8	6	300	RISERS	R	1	12	300	STEAM OUTLET CONNECTION
2 GAS INLET CONE CONNECTIONS.					S	3	3	300	SAFETY VALVE CONNECTION
H	1	24	150	MANWAY (not by others)	T	1	2	300	CONTINUOUS BLOWDOWN
I	3	2	300	THERMOWELL CONNECTION	U	1	3/4	6000	COUPLING FOR CHEMICAL FEED
J	1	3/4	6000	THREADED COUPLING FOR PRESSURE GAUGE CONNECTION	V	2	1½	6000	COUPLING FOR LEVEL GAUGES
					W	2	2	300	LEVEL CONTROL NOZZLE
3 GAS OUTLET, CONE CONNECTIONS					X	1	1½	1000	LOW LEVEL ALARM COUPLING
K	1	24	150	MANWAY (not by others)	Y	1	1½	1000	HIGH LEVEL ALARM CONNECTION
L	3	2	300	THERMOWELL CONNECTIONS	Z	1	6	300	AUXILIARY NOZZLE
M	1	3/4	6000	THREADED COUPLING FOR PRESSURE GAUGE CONNECTIONS	~~AA~~	~~1~~	~~1~~	~~1000~~	[illegible]
					BB	1	3/4	6000	PRESSURE GAUGE CONNECTION
					CC	1	3/4	6000	DRAIN CONNECTION
					DD	1	12"×16"	300	FLUED MANHOLE

STAMPING SPACE

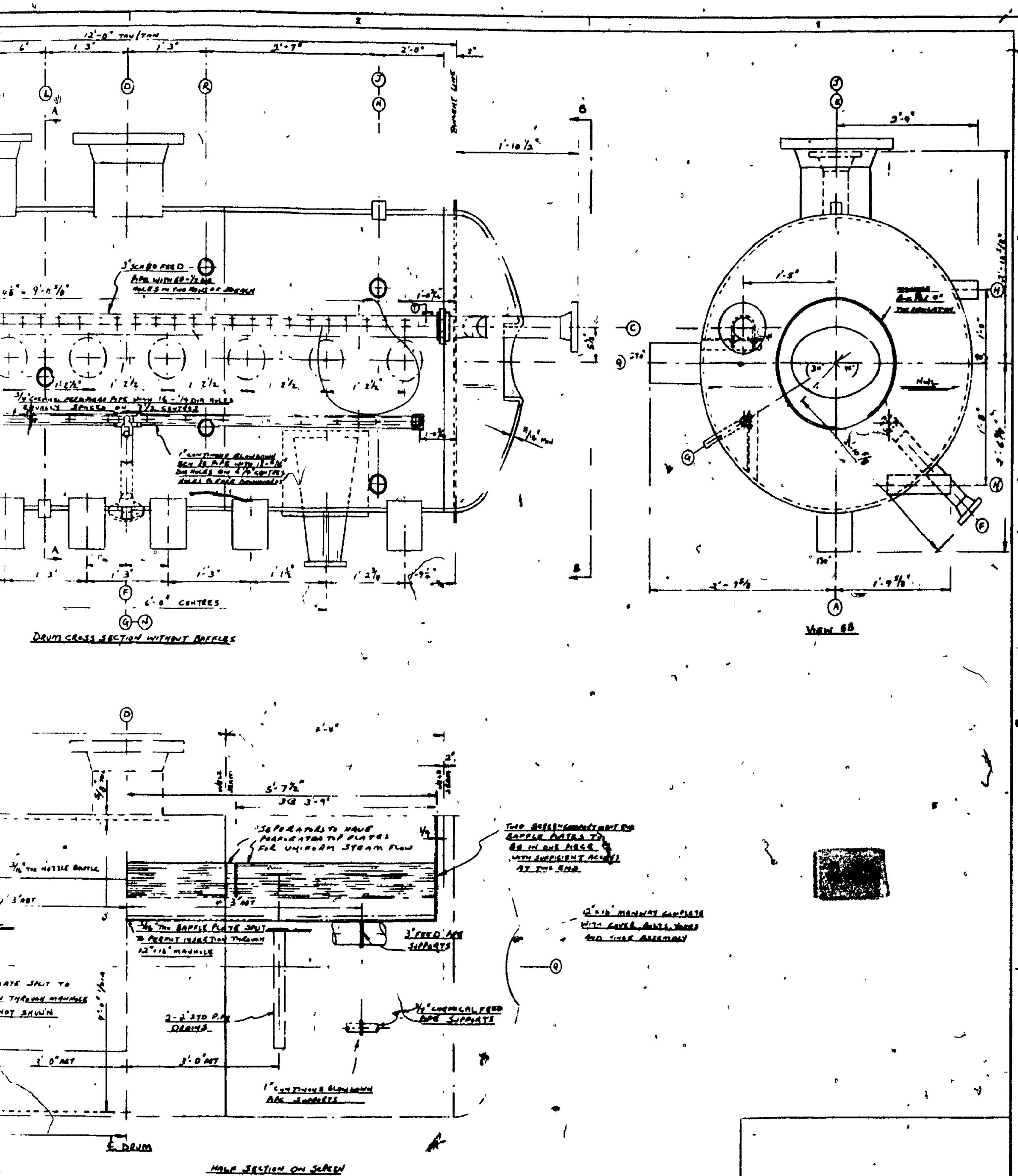
DRAWN	J S KANDOLA
DATE	5-12-78
APPROVED	[illegible]
DATE	13-12-78

Dominion Bridge Company, Limited
ÉQUIPEMENT SUR COMMANDE / ENGINEERED PRODUCTS

DESCRIPTION	ONE FGIM WASTE HEAT EXCHANGER
DETAIL	GENERAL ARRANGEMENT
	IMPERIAL OIL LTD
	E1



P	1	3/4	6000	HALF COUPLING WITH PLUG
R	2	1 1/2	[illegible]	HIGH LEVEL ALARM COUPLINGS
O	1	12"x16"	—	MANWAY
~~P~~	~~1~~	~~1~~	~~6000~~	[illegible]
N	1	3/4	6000	DRAIN CONNECTION
M	1	6	300	AUXILIARY NOZZLE
L	2	1 1/2	6000	LOW LEVEL ALARM COUPLINGS
~~K~~	~~1~~	~~1~~	~~6000~~	[illegible]
J	1	3/4	6000	PRESSURE GAUGE COUPLING
I	2	2	300	LEVEL CONTROL NOZZLES
H	2	1 1/2	TOE	1 1/2 A379 PIPE FOR LEVEL GAUGES
G	1	3/4	6000	COUPLING FOR CHEMICAL FEED
F	1	2	300	CONTINUOUS BLOWDOWN
E	3	2	300	SAFETY VALVE CONNECTION
D	1	12	300	STEAM OUTLET CONNECTION
C	1	3	300	FEED WATER CONNECTION
B	6	6	—	RISER CONNECTIONS
A	6	6	—	[illegible]



HALF SECTION ON SCREEN THRU DRUM VERTICAL ℄

HALF COUPLING WITH PLUG
HIGH LEVEL ALARM COUPLINGS
MANWAY
~~[illegible]~~
DRAIN CONNECTION
AUXILIARY NOZZLE
LOW LEVEL ALARM COUPLINGS
~~[illegible]~~
PRESSURE GAUGE COUPLING
LEVEL CONTROL NOZZLES
1½ XSTD PIPE FOR LEVEL GAUGES

NOTES:-
1. FOR GENERAL NOTES SEE DWG GN-1
2. ALL FLANGES TO BE 300 # RAISED FACE WELDING NECK TO ANSI B 16.5 AND SA 105. ALL INTERNAL THREADED FITTINGS [illegible]
3. ALL NOZZLE NECKS TO BE FABRICATED FROM X-STD PIPE TO SA 106 GR B.
4. X INDICATES FOR INFORMATION ONLY
5. SADDLE DETAILS [illegible] FIG 4 .
6. [illegible] TO BE COMPLETED PRIOR

REMARQUES GÉNÉRALES
GENERAL NOTES

CAHIER DESCRIPTIF / SPECIFICATIONS: ASME SECTION VIII DIV 1 (1977) TEMA (1978)
PIPING TO ANSI B 31.3 (1976)
IBP 5-1-1, IBP 18-6-1, IBP 1-7-1, IBP 20-1-1
IBP 20-2-100

DONNÉES DU DESIGN / DESIGN DATA

PRESSION DU DESIGN / DESIGN PRESSURE: STEAM DRUM 180 PSIG, EXCHANGER 200 PSIG. CONES 40 PSIG. OPERATING PRESSURES: DRUM = 155 PSIG EXCHANGER = 174 PSIG CONES = 15 3 PSIG

TEMPÉRATURE DU DESIGN / DESIGN TEMPERATURE: 650°F
OPERATING TEMPERATURES: STEAM DRUM = 361°F
EXCHANGER = 368°F, INLET CONE REFRACTORY LINED 1400°F

CORROSION ADMISSIBLE / CORROSION ALLOWANCE: 0.125 IN

RENDEMENT DES JOINTS / JOINT EFFICIENCY
PAROI / SHELL: STEAM DRUM AND EXCHANGER AND CONES 100% WITH FULL RT
TÊTES / HEADS: STEAM DRUM = 100%.

LIGAMENT: TUBESHEETS

PRESSION MAXIMALE DE MARCHE À L'ÉTAT DE NEUF ET À FROID
MAXIMUM ALLOW WORKING PRESSURE NEW AND COLD

RESTREINT PAR / LIMITED BY

CAHIER DESCRIPTIF DES MATÉRIAUX / MATERIAL SPECIFICATIONS

	ITEM	MATERIAL	IMPACT VALUES AT CET -20°F	
EXCHANGER	SHELL	SA 516 GR 70	25/20 FT LBF	
	TUBESHEET	SA 516 GR 70	25/20 FT LBF	
	TUBES	SA 192	NONE	
	NOZZLES	SA 106 GR B	NONE	
	HANDHOLES	SA 106 GR B	NONE	
	MANWAY	SA 516 GR 70	NONE	
	MANWAY FLANGES	ANSI B16.5 SA 105	NONE	
	TUBE SUPPORT PLATES	A 36 OR CSA G40 21-44W	NONE	
	SUPPORT PLATES	SA 516 GR 70	25/20 FT LBF	
STEAM DRUM	SHELL	SA 516 GR 70	25/20 FT LBF	
	MANWAY HEAD	SA 516 GR 70	25/20 FT LBF	
	PLAIN HEAD	SA 516 GR 70	20/16 FT LBF	
	NOZZLES	SA 106 GR B	NONE	
	FLANGES	ANSI B16.5 SA 105	NONE	
	INTERNAL PIPE / INTERNAL	A 53 or A 106 / CSA G40 21	NONE	
	SADDLE & ATTACHMENT ANGLE	SA 516 GR 70	25/20 FT LBF	
PIPING	DOWNCOMER RISERS	SA 106 GR B	NONE	
CONES	SHELL	SA 516 GR 70	NONE	
	MANWAY NECKS	SA 516 GR 70	NONE	
	FLANGES	SA 516 GR 70	NONE	
MAIN FLANGES	FLANGES	SA 516 GR 65	NONE	

CHARPY V NOTCH IMPACTING TESTING SHOWN ABOVE APPLIES TO MATERIAL AND DEPOSITED WELD METAL

1 of

CAHIER DESCRIPTIF OU SOUDAGE / WELDING SPECIFICATIONS: EP. WS-981

PROGRAMME D'ASSURANCE-QUALITÉ / QUALITY ASSURANCE: QUALITY CONTROL PROGRAM 981

PROCÉDÉ DE FABRICATION / MANUFACTURING PROCEDURE: 981

SYMBOLES DE SOUDAGE / WELDING SYMBOLS

SYMBOLES D'USINAGE / MACHINING SYMBOLS

TOLÉRANCE CIRCULAIRE / OUT OF ROUN[D]
EXCHANGER SHELL
STEAM DRUM = O
ALL OTHER TOLERANC[E]

INSPECTION

ENREGISTREMENT NO / REGISTRATION NO

PEINTURE / PAINT: EXTERNAL S[...] OF ZINC CROMATE

EXPÉDITION / SHIPPING: EQUIPM[...] FOR SHIPMENT SEE

REMARQUES DIVERSES / MISC NOTES: F[...]
JOINTS SEE EI-3
WASTE HEAT EXCHANG[ER]
SUPPORT STIFFENERS
OTHERS. STEAM DR[UM]
PIPING TO HAVE 4 IN TH[...]
BY OTHERS IN FIEL[D]

HOLES OF ALL FLA[NGES]

DÉTENTE DE CONTRAINTE / STRESS RELIEF
ALL STRESS RELIEF AT 1100-1200°F WITH FOLLOWING HOLD TIMES: TOP TUBESHEET TO HAVE 2 PWHT = 4 HRS TOTAL. WHS SHELL BEFORE TUBING AND CONE SUBSHELLS = 2 1/2 HOURS STEAM DRUM 1-HOUR. FOR DETAILS SEE SKETCHES

RADIOGRAPHIE / RADIOGRAPH
ALL BUTT WELDS ON EXCHANGER AND STEAM DRUM AND CONES TO BE R.T
ALL NOZZLE TO SHELL WELDS TO BE M.T.
5% PIPING WELDS TO BE RT TO ANSI B31.3 (1976)
M.T. AS DETAIL DRAWINGS.

ESSAI HYDROSTATIQUE / HYDROSTATIC TEST
PRESSION / PRESSURE: STEAM DRUM = 324 PSIG (ASME 270 PSIG)
EXCHANGER SHELL SIDE = 360 PSIG (ASME 300 PSIG)
CONE BUTT WELDS R T AND NO SHOP HYDROTEST
TEMPÉRATURE DE L'EAU / WATER TEMP: 60°F
FOR HYDRO-TEST SEE EI-6

3	ADD'L NOTE ADDED, AS NOTED		
2	APPROVAL NOTES ADDED		7/6/79
1	GENERAL REVISION		

UES °F
8F
8F
8F
F
F
8F
8F

...WN ABOVE APPLIES

...TAL

...IG (ASME 270 PSIG)

...G (ASME 300 PSIG)

...YDROTEST

CAHIER DESCRIPTIF DU SOUDAGE
WELDING SPECIFICATIONS
EP. WS-981

PROGRAMME D'ASSURANCE-QUALITÉ
QUALITY ASSURANCE PROGRAM
QUALITY CONTROL PROGRAM 981

PROCÉDÉ DE FABRICATION
MANUFACTURING PROCEDURE
981

SE RÉFÉRER À L'INSTRUCTION DE L'INGÉNIERIE EI 2 (DERNIÈRE RÉVISION)

REFER TO ENGINEERING INSTRUCTION EI 2 (LATEST REVISION)

SYMBOLES DE SOUDAGE
WELDING SYMBOLS
— AMERICAN WELDING SOCIETY

SYMBOLES D'USINAGE
MACHINING SYMBOLS
— DOMINION BRIDGE COMPANY, LIMITED

TOLÉRANCE CIRCULAIRE / OUT OF ROUNDNESS

EXCHANGER SHELL 1.0 IN I.E. 1% OF MEAN DIA.
STEAM DRUM = 0.48 IN " " "
ALL OTHER TOLERANCE TO DB WORKMANSHIP STANDARDS

INSPECTION

ENREGISTREMENT NO / REGISTRATION NO

PEINTURE / PAINT EXTERNAL SURFACES TO BE PAINTED ONE COAT OF ZINC CROMATE SEE SK EI-1

EXPÉDITION / SHIPPING EQUIPMENT TO BE SUITABLY PROTECTED FOR SHIPMENT SEE SK EI-101

REMARQUES DIVERSES / MISC NOTES FOR SOAP TEST OF TUBE TO TUBESHEET JOINTS SEE EI-3
WASTE HEAT EXCHANGER INCLUDING SUPPORT RINGS AND SUPPORT STIFFENERS TO HAVE 4 IN THK INSULATION BY OTHERS. STEAM DRUM TO HAVE 4 IN THK INSULATION. PIPING TO HAVE 4 IN THK INSULATION. ALL INSULATION IS BY OTHERS IN FIELD.

3 HOLES OF ALL FLANGES TO STRADDLE ℄'S

LISTE DES DESSINS
LIST OF DRAWINGS

E1	GENERAL ARRANGEMENT
1	EXCHANGER DETAILS
2	DETAILS OF GAS OUTLET CONE
3	DETAILS OF STEAM DRUM
4	STEAM DRUM INTERNALS
5	MANWAY DAVIT AND BAFFLE PLATES
6	DOWNCOMER PIPING
7	RISER PIPING
8	DETAILS OF GAS INLET CONE
X 1107	MANHOLE DAVIT FOR STEAM DRUM
SK 6	LISTING OF STUD BOLTS AND GASKETS

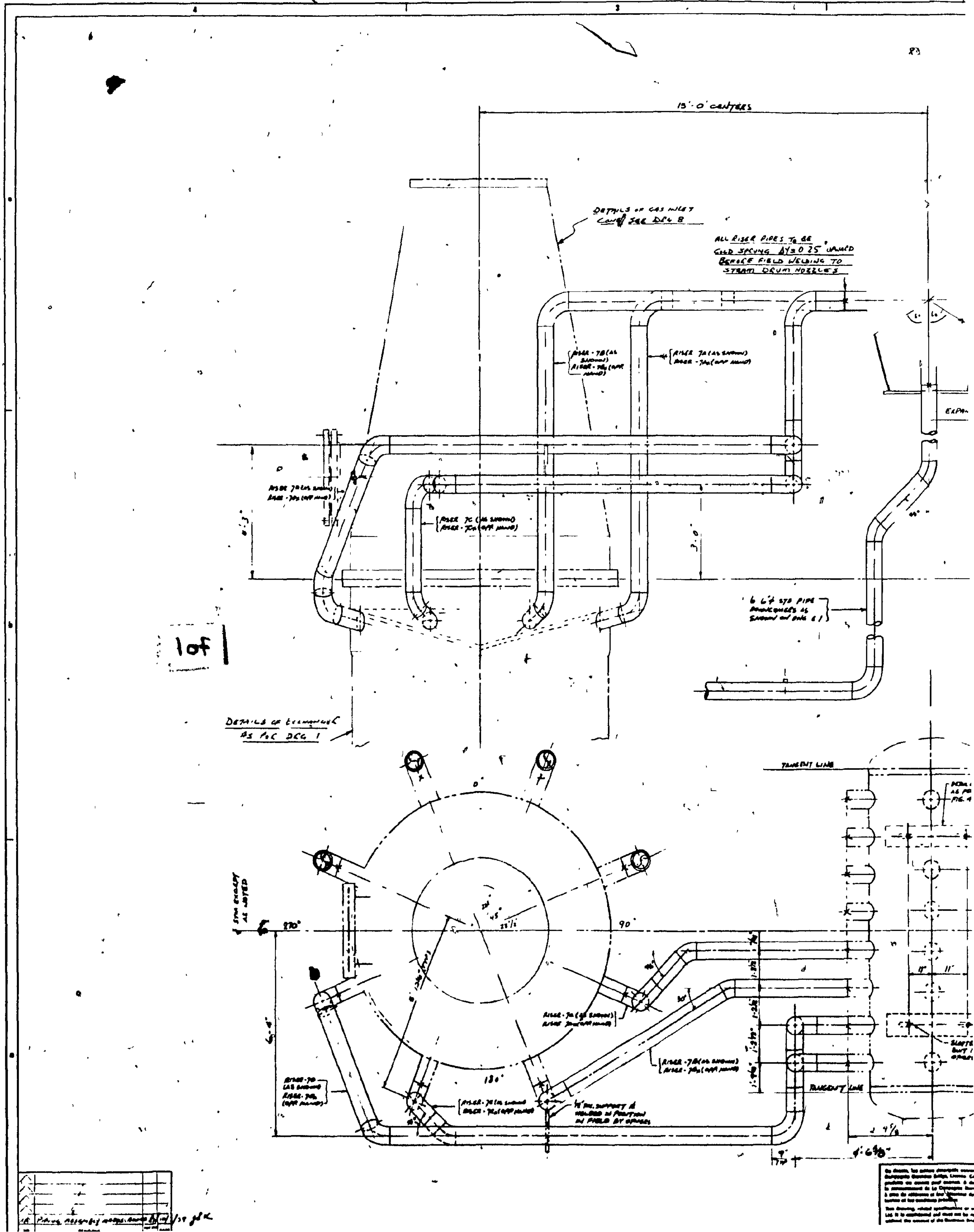
STRESS RELIEF SKETCHES:

EI R 1 :	TOP TUBESHEET
EI R 2 .	EXCHANGER SHELL BEFORE TUBING
EI R 3 :	STEAM DRUM
EI R 9 .	MAIN FLANGES

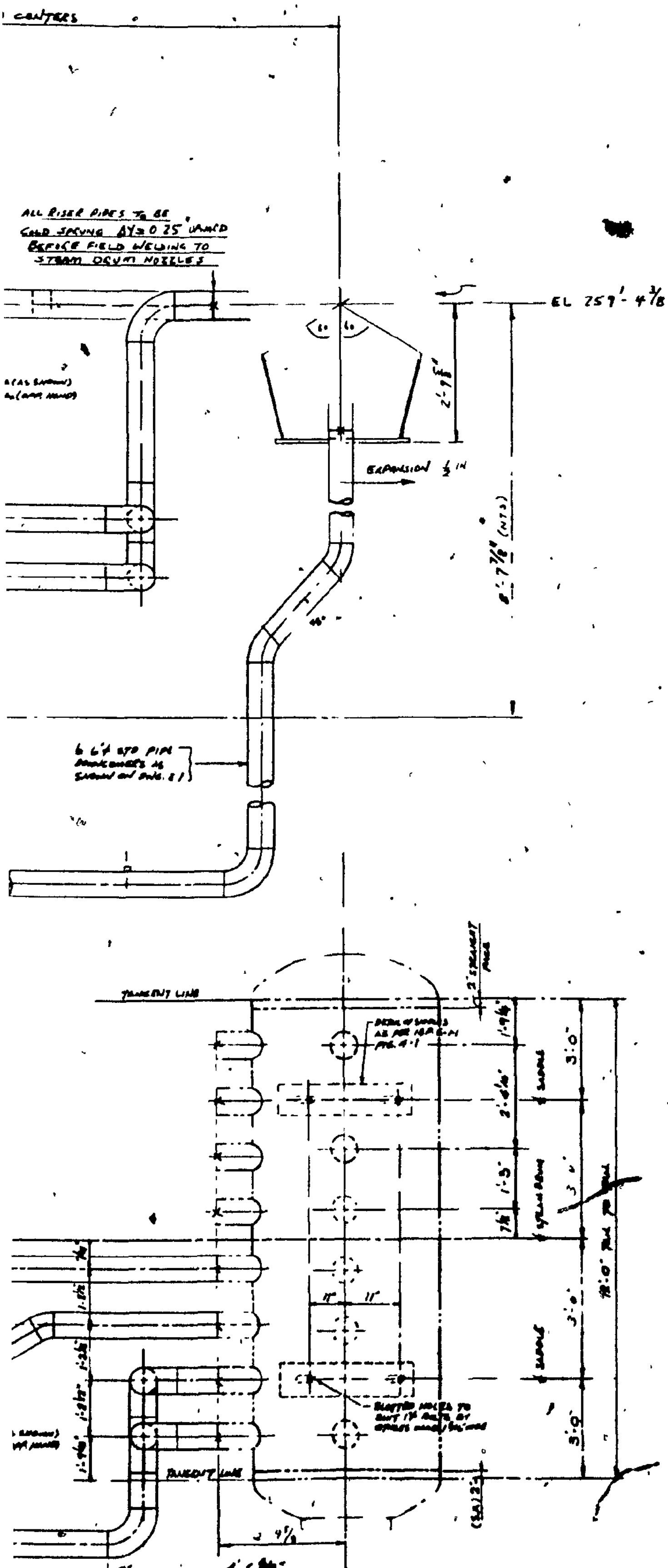
NORMALISING INSTRUCTIONS:-

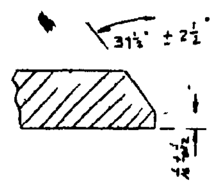
EI R 4	TUBESHEET MATERIAL
EI R 5	MAIN FLANGE MATERIAL
EI R 6	DRUM SHELL MATERIAL
EI-3	TESTING OF TUBESHEET MATERIAL
EI 7	WELD PRODUCTION TEST PLATES
EI 8	TESTING OF STEAM DRUM MATERIAL
EI 9	TESTING OF MAIN FLANGE MATERIAL

DESSINS DE RÉFÉRENCE / REFERENCE DRAWINGS



15'-0" CENTERS
DETAILS OF GAS INLET
CONE SEE DRG 8
ALL RISER PIPES TO BE
COLD SPRUNG ΔY=0.25" UPWARD
BEFORE FIELD WELDING TO
STEAM DRUM NOZZLES
1of1
DETAILS OF EXCHANGER
AS PER DRG 1
TANGENT LINE
0°
90°
180°
270°
TANGENT LINE





TYPICAL WELD PREP DETAIL
FOR PIPE AND FITTINGS
5% WELDS TO BE R.T.
TO ANSI B31.3 1976

ALL PIPING TO BE SA 106 GR B
ALL FITTINGS ELBOWS TO BE ASTM A 234 WPB AND ANSI B16.9
FOR SADDLES USE SA 516 - GR 70 PLATE FROM 5/8" x 94" x 40'-0" ALREADY RESERVED FROM NON STOCK S I SEE CUTTING SK 4
—X— INDICATES FIELD WELDS BY OTHERS

DEADWEIGHT
DRUM EMPTY 7 KIP — PIPING EMPTY 11 K
DRUM FLOOD 12 KIP — PIPING FLOODED 18.5 K

TOTAL WEIGHT 36.5 KIP
LOAD AT EACH SADDLE = 19 KIP EXCLUDING INSULATION AND MOUNTINGS BY OTHERS

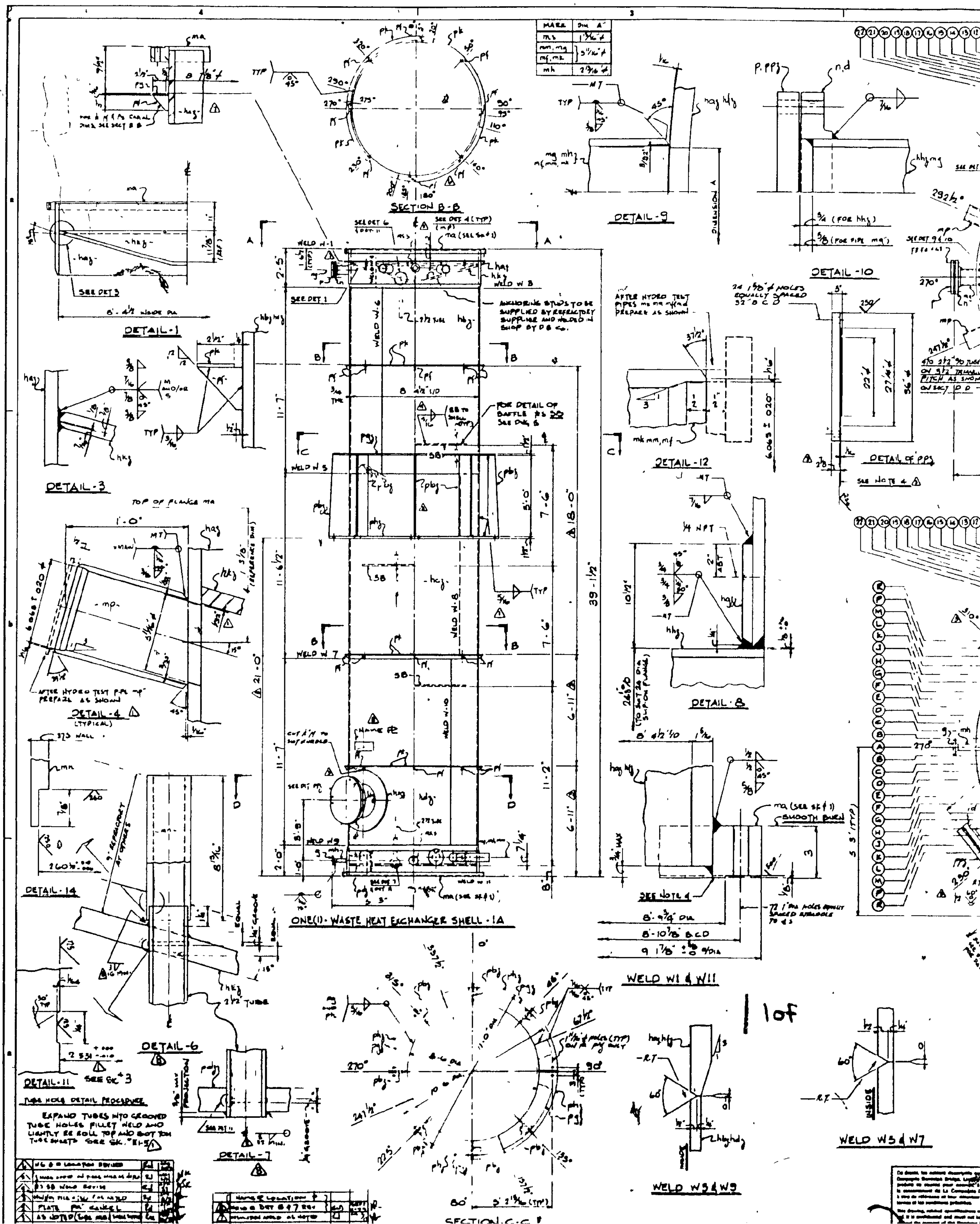
FOR GENERAL NOTES SEE DRG GN 1

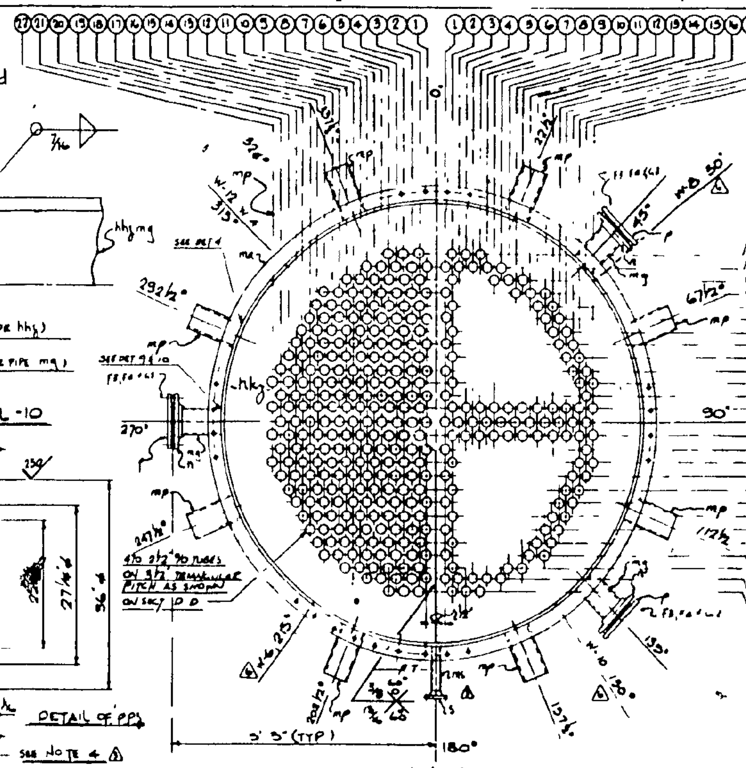
Dominion Bridge Company, Limited
ÉQUIPEMENT SUR COMMANDE / ENGINEERED PRODUCTS

J S Kandola 9 March 77

DESCRIPTION: ONE (1) FCIM WASTE HEAT EXCHANGER
DETAIL: RISER AND DOWNCOMER PIPING LAYOUT

E101 / A

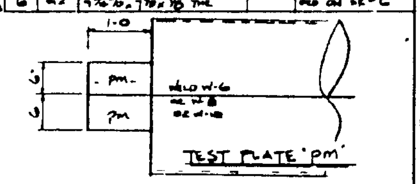

SECTION B-B
DETAIL-1
DETAIL-3
DETAIL-4
DETAIL-6
DETAIL-7
DETAIL-8
DETAIL-9
DETAIL-10
DETAIL-11
DETAIL-12
DETAIL-14
DETAIL OF PPS
ONE(1)- WASTE HEAT EXCHANGER SHELL - 1A
WELD W1 & W11
WELD W3 & W9
WELD W5 & W7
SECTION C-C
ANCHORING STUDS TO BE SUPPLIED BY REFRACTORY SUPPLIER AND WELDED IN SHOP BY DB Co.
AFTER HYDRO TEST PIPES mk mm mf and PREPARE AS SHOWN
24 1 5/8" ϕ HOLES EQUALLY SPACED 32" B C D
FOR DETAIL OF BAFFLE P's 20 SEE DWG 6
TUBE HOLE DETAIL PROCEDURE
EXPAND TUBES INTO GROOVED TUBE HOLES FILLET WELD AND LIGHTLY RE ROLL TOP AND BOT TOM TUBE SHEETS SEE SK. "E1-5"
1 of



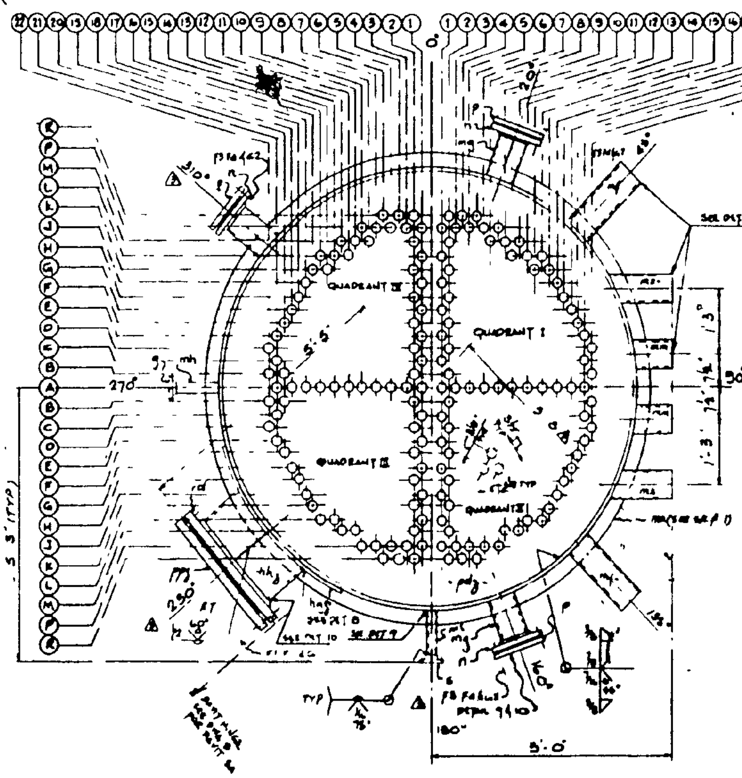
PLAN VIEW A-A

QTY	MARK	DESCRIPTION	LENGTH	MATERIALS
1	ha	PL 1 3/16 x 28 1/4	36' 7 3/8	SA 516 GR 70
1	hb	PL 3/4 x 139	36-6 1/4	-
1	hc	PL 3/4 x 138 1/2	36 6 1/4	-
1	hd	PL 3/4 x 139	36 6 1/4	-
1	hf	PL 1 3/16 x 23 1/4	36 7 1/8	-
1	hg	PL 3/4 x 45	4' 0"	-
1	hh	PL 1/2 x 15 1/16	6 1 3/4	-
		~~[illegible]~~		
8	pb	PL 1 3/16 x 15	5-0"	
3	5A	BAFFLE PLS		ORD ON DWG 5
1	pd	100 7/8 DIA x 1 3/16 THK PL		SA 516 GR 70
32	pf	PL 4 x 3	3"	-
4	pg	PL 1/2 x 84 (CUT 4 pg)	7-9"	-
4	ph	PL 1/2 x 96 (CUT 4 ph)	8 0"	-
12	pk	PL 3/16 x 35 1/2 (CUT 8 pk)	6 4 1/2"	A240 GR 304
4	pj	PL 3/16 x 34 (CUT 4 pj)	6' 5"	
2	ma	FLANGE RING		ORD ON SK #1
470		2 1/2 OD x .083 WALL TUBES		SA-192
1	hk	106 9/32 DIA x 1 3/16 THK PL	—	SA 516 GR 70
2	mf	6" x .432 WALL (X-STG) PIPE	4 7/8"	SA 106 GR B
6	mg	6" x .432 WALL (X-STG) PIPE	11 3/4"	-
1	mh	3" x .300 WALL (X-STG) PIPE	8 3/4"	-
2	mk	6" x .432 WALL (X-STG) PIPE	1 3/4"	-
2	mm	6" x .432 WALL (X-STG) PIPE	9 1/2"	-
470	mn	2 1/2" x .134 WALL PIPE	8	SS TYPE 410
8	mp	6" x .432 WALL (X-STG) PIPE	1 1/16"	SA 106 GR B
2	ms	2" x .218 WALL (X-STG) PIPE	6 3/4"	SA 106 GR B
2	pm	TEST PL 3/4 x 6	1-0	SA 516 GR 70
1	d	36" 300# SLIP ON FLANGE		SA 105
1	ppj	PL 3" x 36 5/8		SA 516 GR 70
1	g	36" 300# WELDING NECK FLANGE		SA 105
1	5A	OMIT PER 24MH		ORD ON DWG 5
6	n	6" 300# SLIP ON FLANGE		SA 105
6	p	6" 300# BLIND FLANGE		SA 105
2	s	2" 300# BLIND FLG (BORE FOR 2" x [illegible])		SA-105
24	F1	1 1/2" STUD BOLTS	9"	ORD ON SK #6
48	F2	1 1/2" HEX NUTS		-
72	F3	3/4" STUD BOLTS	4 3/4"	-
144	F4	3/4" HEX NUTS		-
1	G1	30 5/8 OD x 24 7/8 ID x 1/8 THK		-
6	G2	9 3/4 OD x 7 5/8 ID x 1/8 THK		ORD ON SK #6

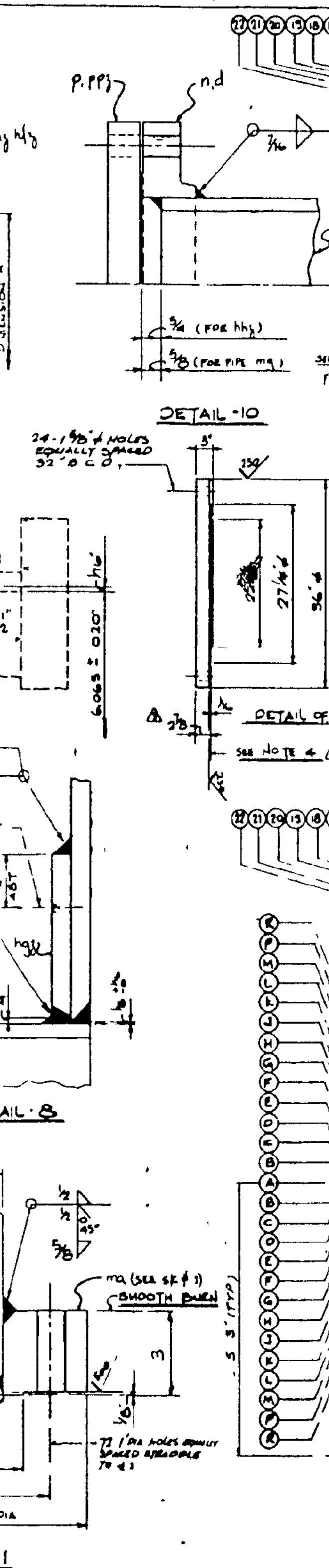
MATERIAL FOR ONE (1) - WASTE HEAT EXCHANGER



TEST PLATE "pm"



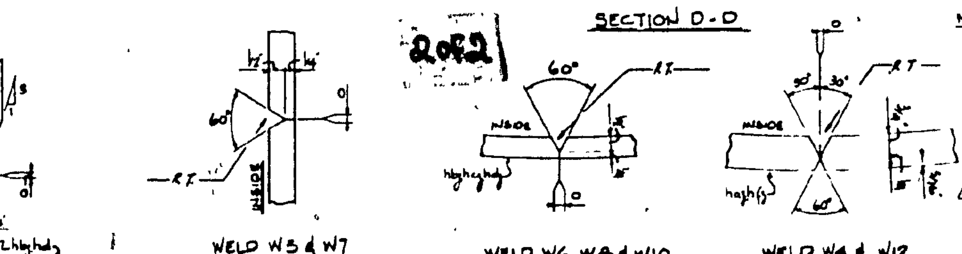
SECTION D-D



DETAIL -10

DETAIL OF ppj

DETAIL - 8



WELD W5 & W7

WELD W6, W8 & W10

WELD W4 & W12

- THEORETICAL CUTTING DIMENSIONS -

TUBE MARK	TUBE LENGTH	TUBE MARK	TUBE LENGTH	TUBE MARK	TUBE LENGTH	TUBE MARK	TUBE LENGTH	TUBE MARK	TUBE LENGTH
A-1	36' 8 1/4"	C-7	36' 10 5/8"	E-13	37' 2 1/16"	H 2	37' 0 3/4"	K 16	37' 5 3/4"
A 3	36' 8 3/16"	C-9	36' 11 3/8"	E-15	37' 2 15/16"	H 4	37' 1 1/16"	L 1	37'-3 1/8"
A 5	36' 9 1/2"	C 11	37' 0 1/16"	E-17	37' 3 3/4"	H-6	37' 1 3/16"	L 3	37' 3 1/4"
A 7	36'-10 1/16"	C 13	37' 1 1/2"	E-19	37'-4 3/8"	H 8	37'-1 7/8"	L 5	37' 3 1/2"
A 9	36' 11 1/8"	C 15	37'-2 1/8"	E 21	37' 5 1/16"	H-10	37'-2 1/2"	L 7	37'-3 5/16"
A 11	37' 0 3/8"	C 17	37' 3 1/8"	F 2	36'-11 3/16"	H-12	37'-3 1/8"	L-9	37'-4 1/4"
A 13	37' 1 1/4"	C 19	37' 4 1/4"	F 4	36' 11 5/16"	H-14	37'-3 5/16"	L 11	37' 4 [illegible]
A 15	37' 2 1/4"	C 21	37' 5 1/8"	F 6	37' 0"	H-16	37' 4 1/8"	L 13	37'-5 1/4"
A 17	37' 3 3/16"	D 2	36' 9 3/8"	F 8	37'-0 7/8"	H 18	37' 5 3/16"	L 15	37' 5 1/8"
A 19	37' 4 1/16"	D 4	36' 10 1/8"	F-10	37' 1 3/16"	J 1	37' 1 1/2"	M-2	37' 4"
A 21	37' 5"	D 6	36' 10 5/8"	F 12	37'-2 1/8"	J 3	37' 1 3/16"	M-4	37' 4 1/8"
B 2	36' 8 7/8"	D 8	36' 11 3/8"	F 14	37'-2 5/8"	J 5	37'-2"	M 6	37'-4 3/16"
B 4	36' 9 1/4"	D 10	37' 0 1/2"	F 16	37' 3 5/16"	J-7	37'-2 3/8"	M 8	37'-4 [illegible]
B-6	36' 10 1/8"	D 12	37' 1 5/16"	F-18	37' 4 1/2"	J-9	37'-3 1/8"	M 10	37'-5 3/16"
B-8	36' 11"	D-14	37'-2 1/16"	F-20	37'-5 3/8"	J-11	37' 3 3/8"	M 12	37' 5 5/16"
B-10	37' 0"	D 16	37'-3 1/16"	G-1	36'-11 3/8"	J 13	37' 4"	P-1	37'-4 1/16"
B-12	37'-0 1/8"	D 18	37' 4"	G 3	37' 0 1/16"	J-15	37'-4 3/8"	P 3	37' 4 1/8"
B-14	37' 1 1/16"	D 20	37'-4 1/8"	G 5	37' 0 5/8"	J-17	37'-5 1/8"	P 5	37'-5"
B 16	37' 2 1/4"	D 22	37' 5 1/8"	G-7	37' 1"	K-2	37'-2 3/8"	P 7	37'-5 [illegible]
B-18	37'-3 1/16"	E 1	36' 10 1/8"	G 9	37' 1 [illegible]	K-4	37'-2 3/8"	P 9	37'-5 [illegible]
B 20	37'-4 1/16"	E-3	36'-0 7/8"	G 11	37' 2 1/4"	K-6	37' 1 1/8"	R-2	37'-5 3/8"
B-22	37' 5 1/2"	E 5	36' 11 1/8"	G 13	37' 2 3/8"	K-8	37' 3 1/2"	R-4	37' 5 1/4"
C-1	36' 8 3/4"	E 7	36' 11 3/4"	G-15	37'-3 1/16"	K 10	37' 3 3/4"	R-6	37'-6"
C 3	36' 9 1/4"	E-9	37'-0 1/2"	G-17	37'-4 1/8"	K-12	37' 4 1/8"		
C 5	36' 10"	E 11	37'-1 1/4"	G 19	37'-5 3/16"	K-14	37' 4 3/16"		

STAMPING SPACE

NOTE.

1 SHOP TO CHECK OF THE ACTUAL LENGTH OF TUBES BEFORE CUTTING (POSITION OF THE TUBE SHEETS)

2 FOR GENERAL NOTES SEE DWG. GN1

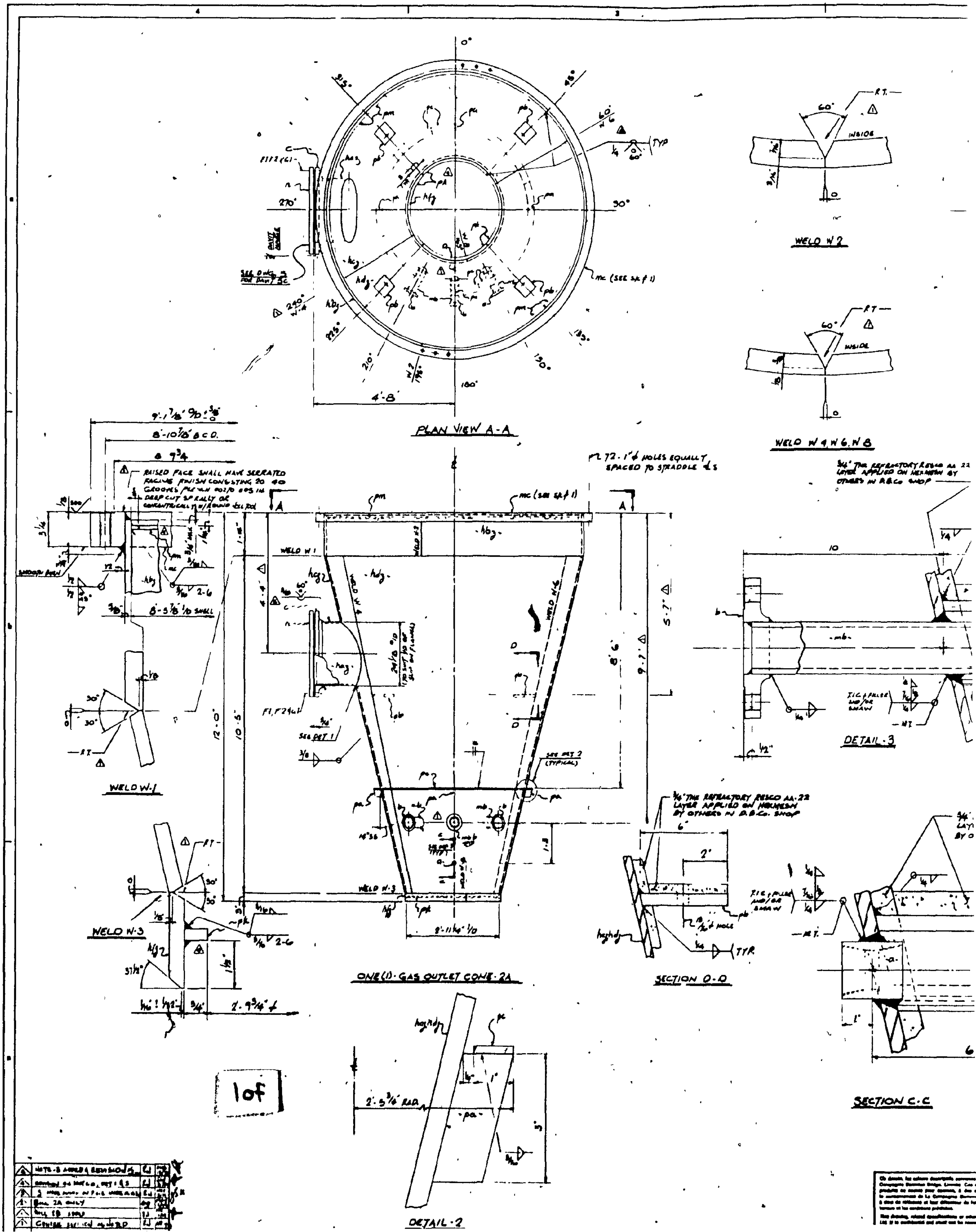
3 FOR TUBE ROLLING SEE SK E15

4 RAISED FACE SHALL HAVE SERRATED FACING FINISH CONSISTING OF 20-40 GROOVES PER INCH .003 TO .006 IN. DEEP CUT SPIRALLY OR CONCENTRICALLY WITH ROUND NOSE TOOL

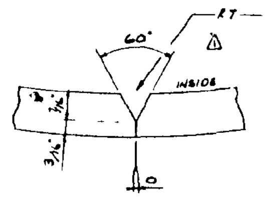
DRAWN	E NAPISA FEB 6/75
CHECKED	GD FEB/16/7?
APPROVED	[illegible]
SECTION	BOILER (435)
	A-B
	Nelle Goode

Dominion Bridge Company, Limited
ÉQUIPEMENT SUR COMMANDE / ENGINEERED PRODUCTS

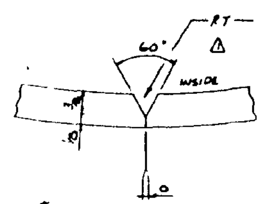
DESCRIPTION: ONE (1) [illegible] WASTE HEAT EXCHANGER
DETAIL: WASTE HEAT EXCHANGER SHELL & TUBES
CLIENT: IMPERIAL OIL LTD.
SHEET: 1 / REV: B
565 - 981



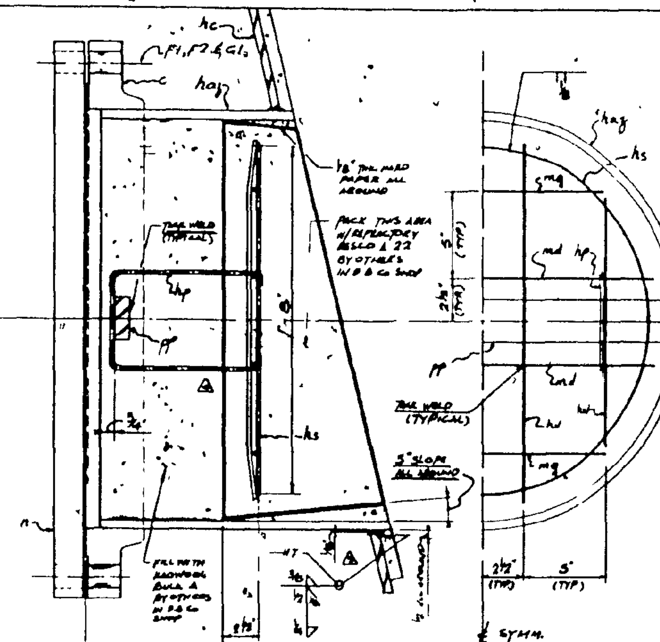
PLAN VIEW A-A
WELD W 2
WELD W 4, W 6, W 8
RAISED FACE SHALL HAVE SERRATED FACING FINISH CONSISTING 30-40 GROOVES PER INCH
WELD W-1
WELD W-3
ONE(1)-GAS OUTLET CONE-2A
SECTION D-D
DETAIL-3
SECTION C-C
DETAIL-2
1of



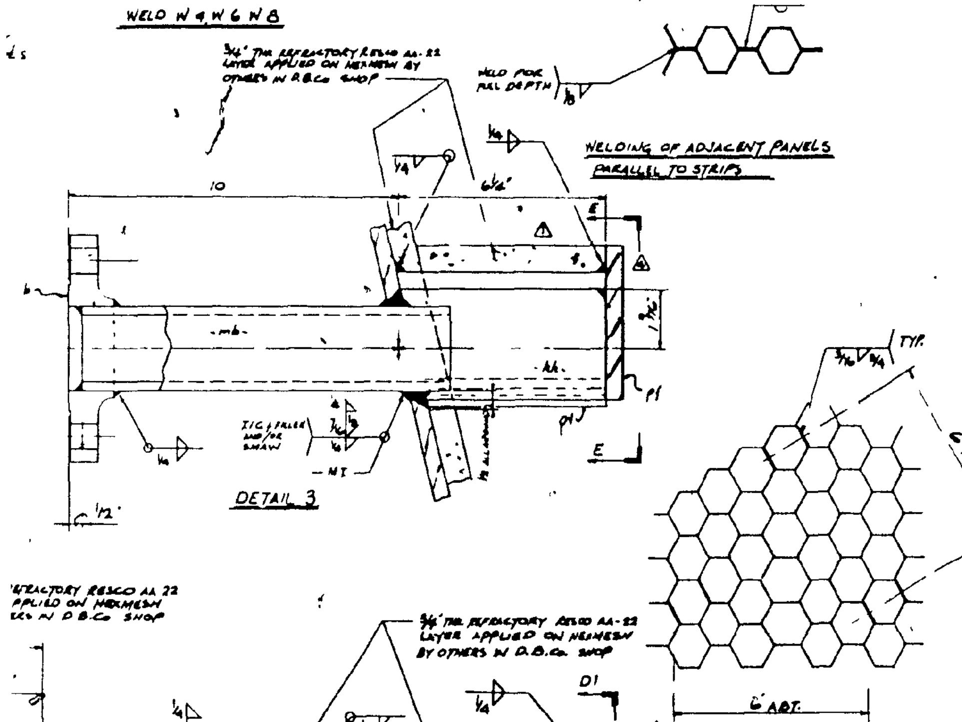
WELD W 2



WELD W 4, W 6, W 8

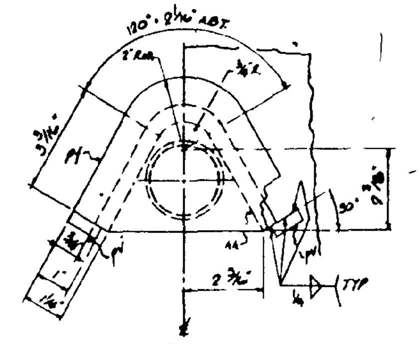


DETAIL - 1



WELDING OF ADJACENT PANELS PARALLEL TO STRIPS

DETAIL 3



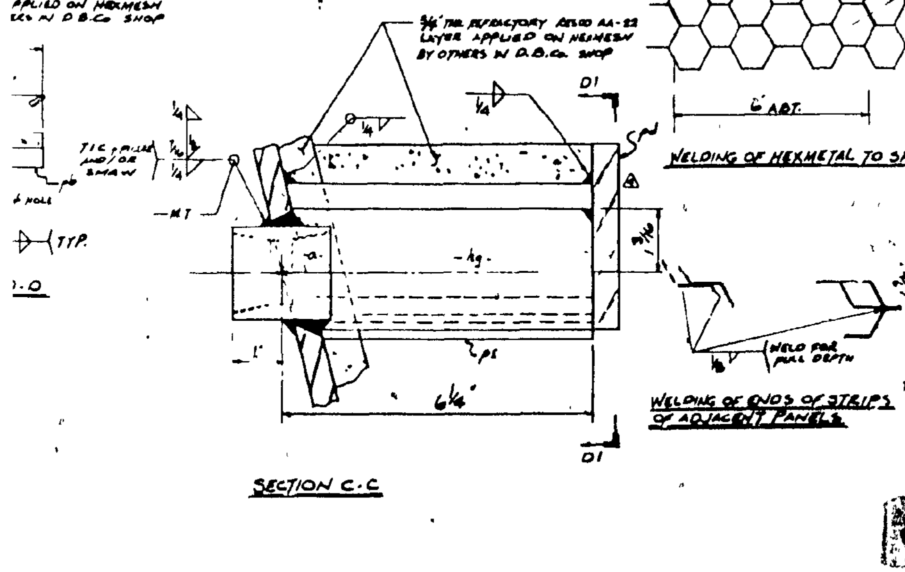
SECTION E-E

NOTES

1 FOR GENERAL NOTES SEE DWG GN1

2 THE TERMINATION OF REFRACTORY AND HEXMESH TO BE AT VAPOR STOPS PIECE MARKS "pk" & "pm"

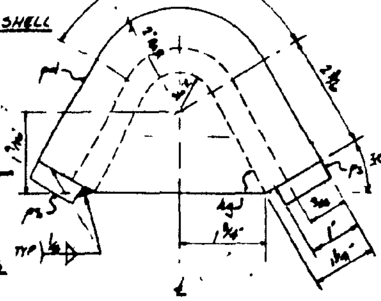
3 HOLES TO ALL FLANGES TO STRADDLE ℄S.



WELDING OF HEXMETAL TO SHELL

WELDING OF ENDS OF STRIPS OF ADJACENT PANELS

SECTION C-C



SECTION D1-D1

MATERIAL FOR ONE(1) - WASTE HEAT EXCHANGER				
QTY	MARK	DESCRIPTION	LENGTH	MATERIAL
1	mc	FLANGE RING		DED on SK 01
3	mb	2"⌀ x .218 WALL (3 3/4) PIPE	11 3/4"	SA-106 GR B
1	SC	PAINT		DED on DWG S
1	hag	ℙ 1/2 x 17 7/8	6' 2 11/16"	SA 516 GR 70
1	hb	ℙ 3/8 x 18 1/4	26' - 10"	"
1	hc	ℙ 1/2 x 137	23' - 6"	"
1	hd	(CUT 1-hc & 1-hd)		"
1	hf	ℙ 1/2 x 3	9' - 4 5/16"	"
1	a	3/4 ⌀ 6000# [illegible]	—	SA 105
3	b	7⌀ 300# SLIP ON FLANGE	—	SA-105
1	c	24⌀ 150# SLIP ON FLANGE	—	SA 105
		3/4 HEXMETAL		S.S TYPE 410
4	pa	ℙ 1/2 x 1 3/4	5 1/2"	SA 516 GR 70
1	n	24" 150# BLIND FLANGE		SA-105
4	pb	ℙ 1 x 6	6"	SA 516 GR 70
4	pc	ℙ 3/16 x [illegible] (CUT 4 pc)	3' - 8"	G40.21 GR [illegible]
1	pd	ℙ 1/2 x 3 1/16	5 3/4"	[illegible]
3	pf	ℙ 1/2 x 4 3/8	6 [illegible]	[illegible]
~~[illegible]~~	~~[illegible]~~	~~[illegible]~~	~~[illegible]~~	~~[illegible]~~
20	F1	1 1/16" ⌀ STUD BOLTS	7 1/16"	[illegible]
40	F2	1 1/16" ⌀ HEX NUTS		[illegible] SK 0[illegible]
1	G1	1/8" THK GASKET		DED on SK 16
1	hg	ℙ 1/2 x 6 1/8	6 5/16"	G40.21 GR [illegible]
3	hh	ℙ 1/2 x 6 3/8	8 1/16"	G40.21 GR [illegible]
~~[illegible]~~	~~[illegible]~~	~~[illegible]~~	~~[illegible]~~	~~[illegible]~~
~~[illegible]~~	~~[illegible]~~	~~[illegible]~~	~~[illegible]~~	~~[illegible]~~
4	pk	ℙ 3/16 x 12 1/8 (CUT 4 pk)	2' - 1"	G40 21 GR [illegible]
4	pm	ℙ 3/16 x 23 (CUT 4 pm)	6' - 0"	"
1	pp	ℙ 3/4 x 3 1/2	1' - 11"	"
2	hp	1/2" ROUND BAR	2' - 4" NET	A 36
1	hs	1/4" ROUND BAR	5' - 7 3/8"	"
2	hu	1/4" " "	1' 3 1/4"	"
2	hv	1/4" " "	1' - 9 1/2"	"
2	mg	1/4" " "	1' 3"	"
2	md	1/4" " "	1' - 9"	"
2	ps	ℙ 3/16 x 1	3 3/8"	G40 21 GR [illegible]
6	pl	ℙ 3/16 x 1	3 1/2"	"

STAMPING SPACE

DRAWN / DESSINÉ: R. NAPISA — DATE: FEB 9/78
CHECKED / VÉRIFIÉ: GND — FEB 6/15/79

APPROVED / APPROUVÉ

Fabriqué au Canada / MADE IN CANADA

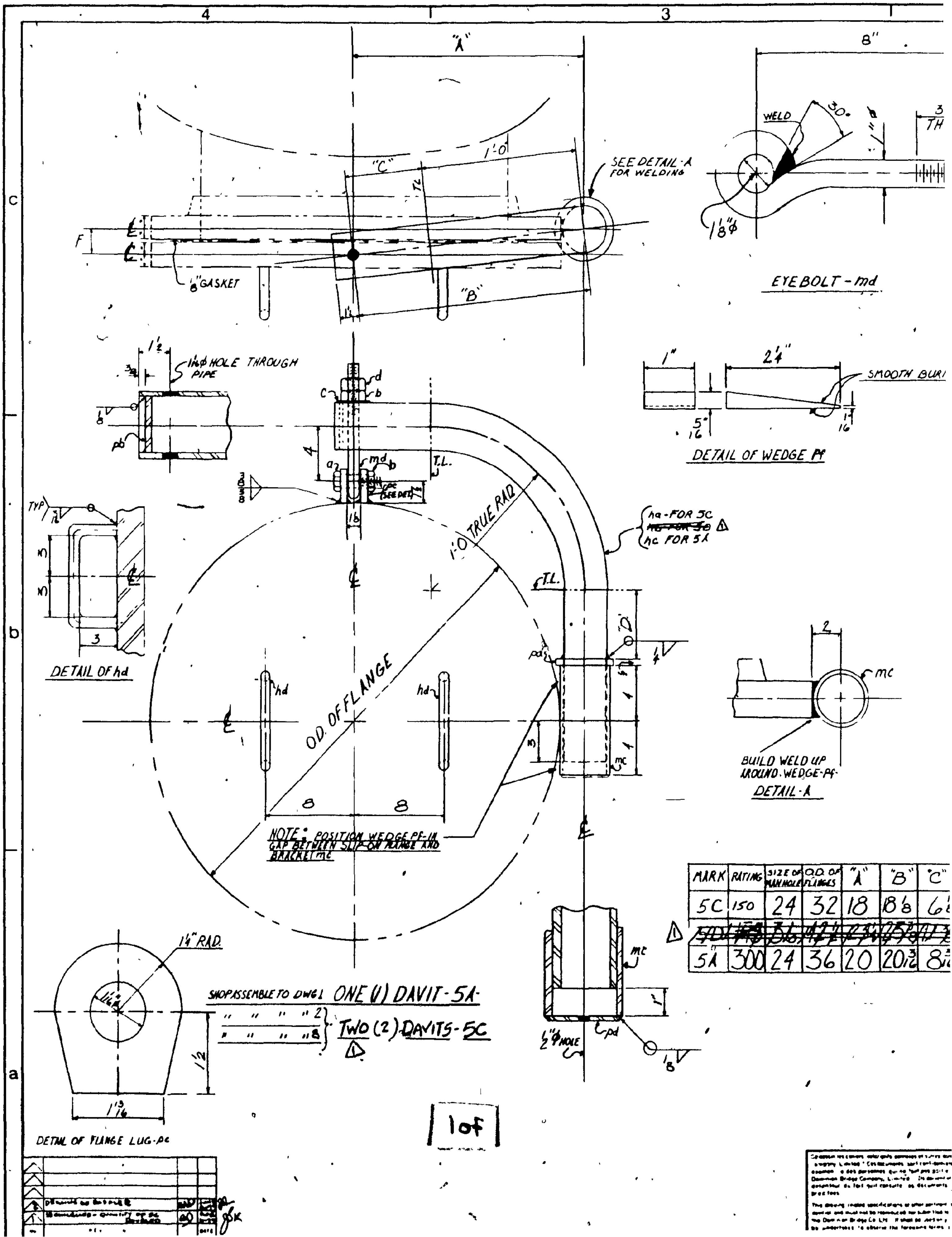
Dominion Bridge Company, Limited.

ÉQUIPEMENT SUR COMMANDE / ENGINEERED PRODUCTS

DESCRIPTION: ONE(1) - FCIM WASTE HEAT EXCHANGER

DETAIL: GAS OUTLET CONE

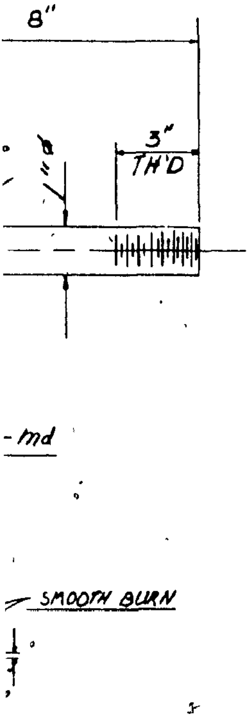
2 — REV. 5

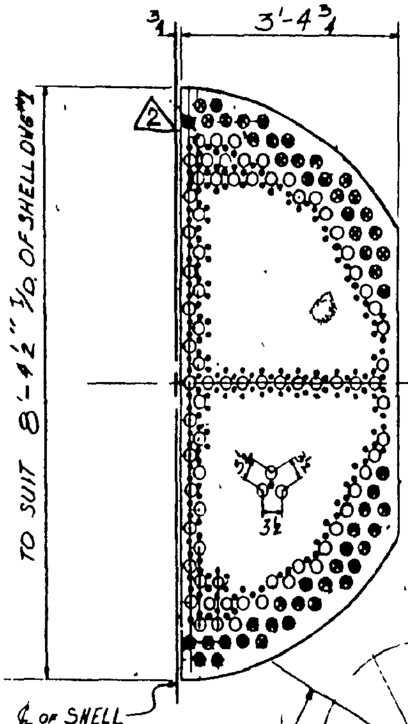


MARK	RATING	SIZE OF MANHOLE	O.D. OF FLANGES	"A"	"B"	"C"
5C	150	24	32	18	18 1/8	6
~~[illegible]~~						
5A	300	24	36	20	20 3/16	8

2

1





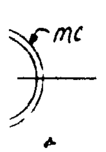
(WORK WITH DWG #1 FOR TUBE DRILLING)

3 BAFFLE PL'S-5B-

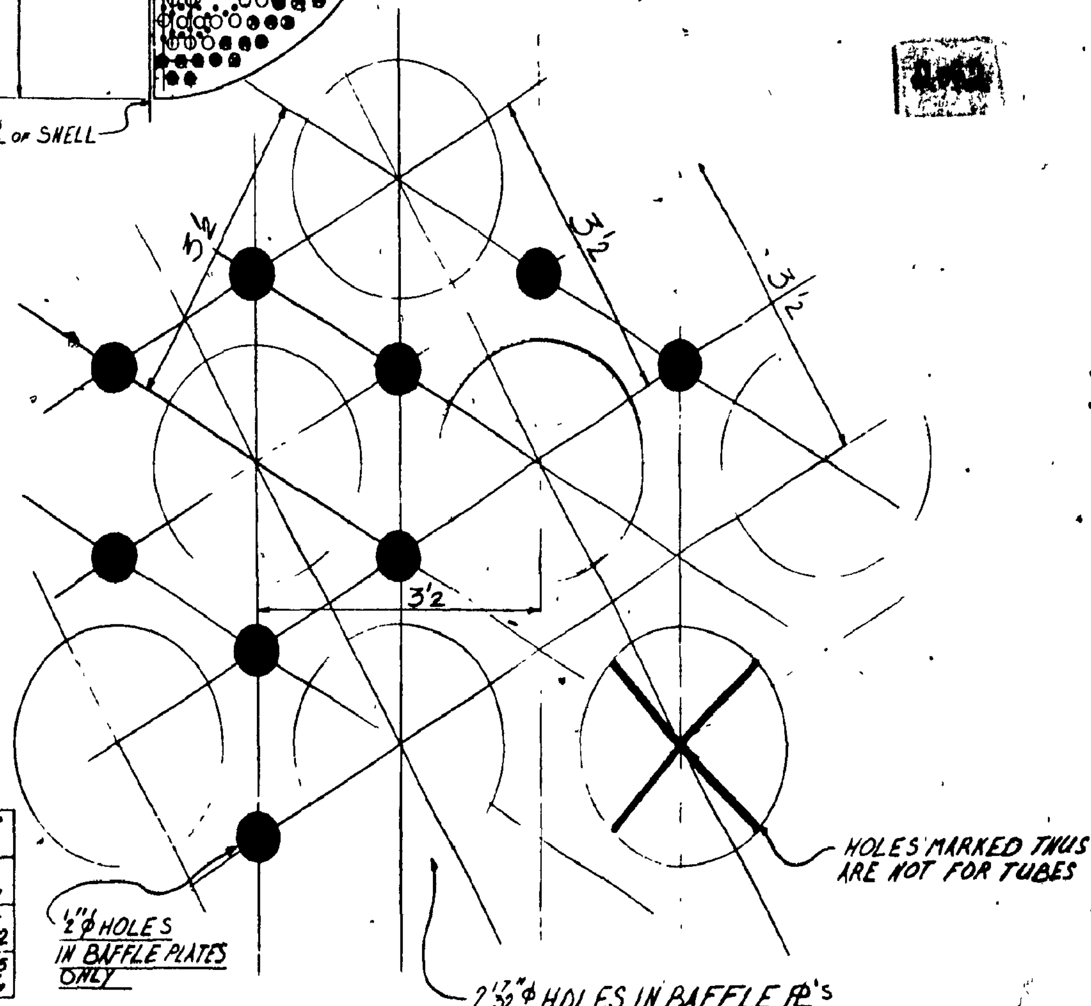
SHOP ASSEMBLE TO DWG #1

MATERIAL FOR ONE (1) WASTE HEAT EXCHANGER				
QTY.	MARK	DESCRIPTION	LENGTH	MATERIAL
3	mc	PI-3 1/2 SCH40 .226 WALL	8"	SA-106 GR B
2	ha	PI-3 SCH.80	3'-3"	A53
~~1~~	~~hb~~	~~PI-3 SCH.80~~	~~[illegible]~~	
1	hc	PI-3 SCH.80	3'-7 1/16"	A53
6	hd	3/4" ϕ ROD	1'-2" ABT	G40.21 44W
3	md	1" ϕ (THREADED 2 1/2" EACH END) ROD	1'-1 ABT	
3	Pd	PL 1/4" X 3 3/4" O.D.	–	
6	Pf	PL 5/16" X 1"	2 1/4"	
3	Pa	PL 1/2" X 4 1/2" O.D. X 3 1/16 I.D.	–	
3	Pb	PL 1/4" X 2 7/8 O/D.	–	
6	Pc	PL 3/4 X 2 1/2	2 3/4"	
3	5B	PL 3/4 X 40 3/4	8'-4 1/2"	
3	a	1" ϕ HEX. CAPSCREW	4"	FA4321E
6	b	1" ϕ HEX NUT		FD5B22V
3	c	1" ϕ WASHER		FE2E22V
3	d	1" ϕ HEX JAM NUT		FD6B22V

c



b



DETAIL OF A SECTION OF BAFFLE PLATE-5B-

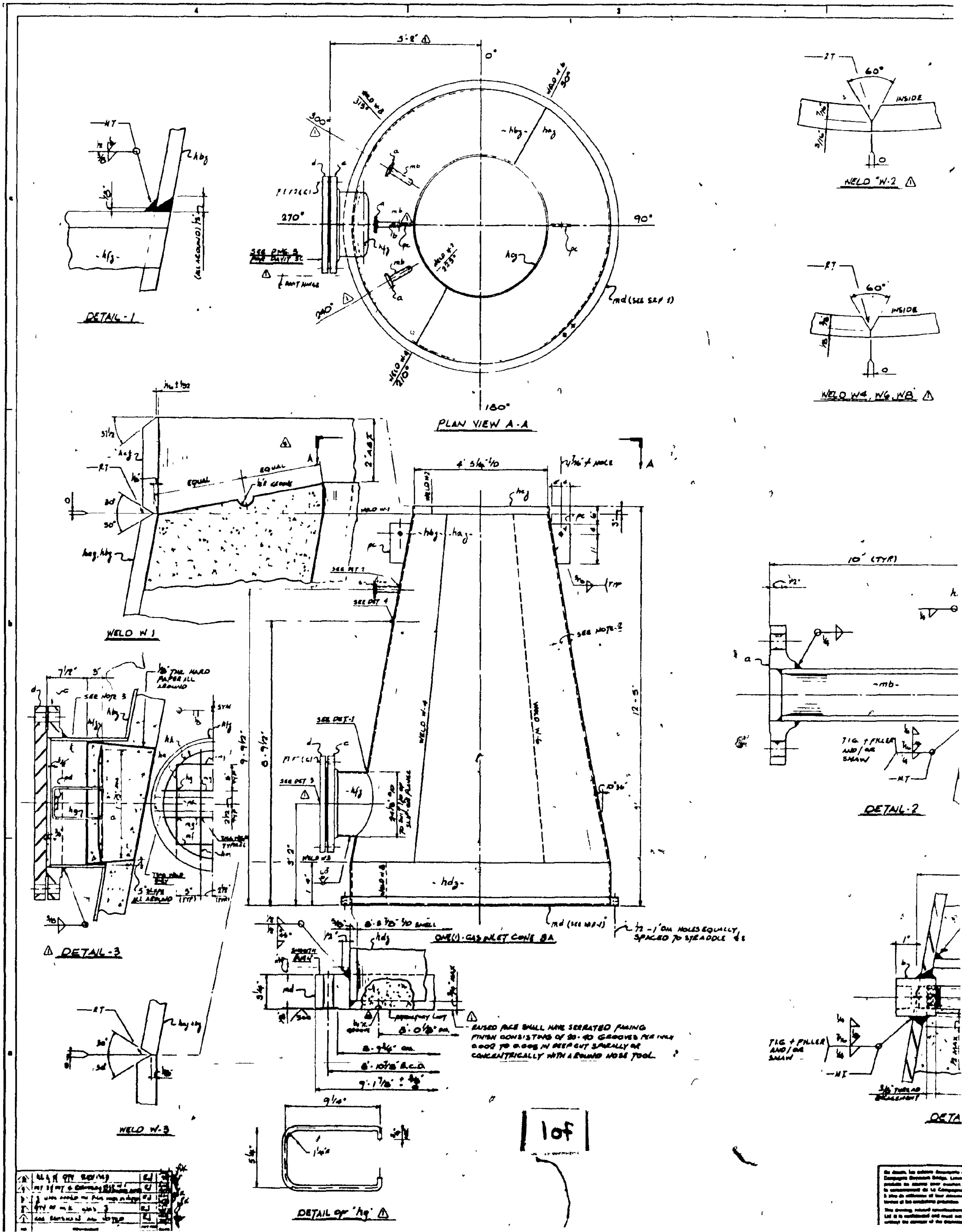
"A"	"B"	"C"	"D"	"F"
18	18 1/8	6 1/8	5	2 1/16
~~[illegible]~~	~~[illegible]~~	~~[illegible]~~	~~[illegible]~~	~~2 1/32~~
20	20 3/16	8 3/16	7	2 15/16

NOTE:

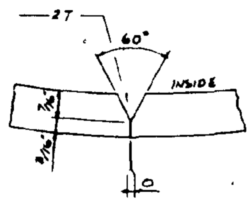
BURRS TO BE REMOVED AROUND TUBE HOLES

a

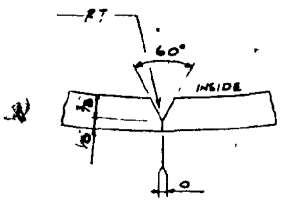
WORK THIS DWG. WITH DWG. 1, 2 & 8



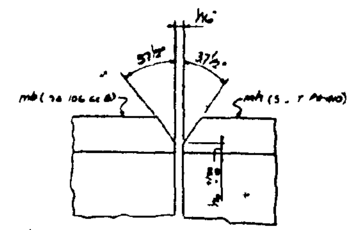
DETAIL-1
PLAN VIEW A-A
WELD "W-2
WELD W4, W6, W8
WELD W1
DETAIL-2
DETAIL-3
WELD W-3
ONE(1)-GAS INLET CONE 8A
DETAIL OF "hg"
RAISED FACE SHALL HAVE SERRATED FACING FINISH CONSISTING OF 30-40 GROOVES PER INCH 0.002 TO 0.005 IN DEEP CUT SPIRALLY OR CONCENTRICALLY WITH A ROUND NOSE TOOL.
1/2-1" DIA HOLES EQUALLY SPACED TO STRADDLE
SEE NOTE 2
SEE NOTE 3
SEE DET-1
SEE DET 3
SEE DET 4
WELD W-4
WELD W3
0°
90°
180°
270°
60°
INSIDE
10" (TYP)
TIG + FILLER AND/OR SMAW
1of



WELD W-2

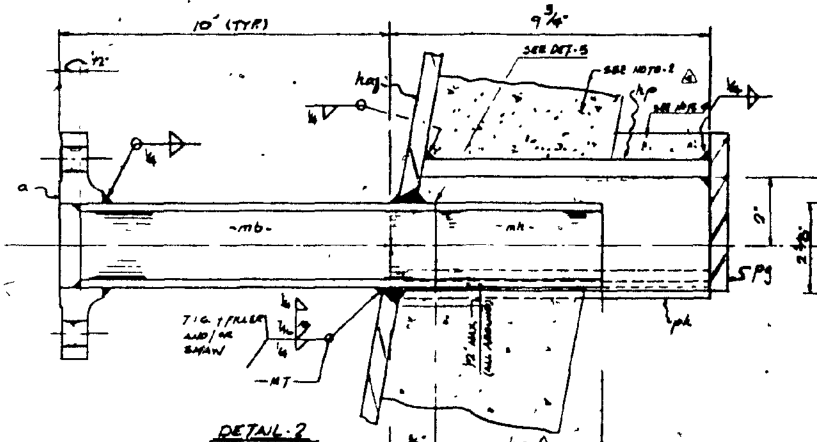


WELD W4, W6, W8

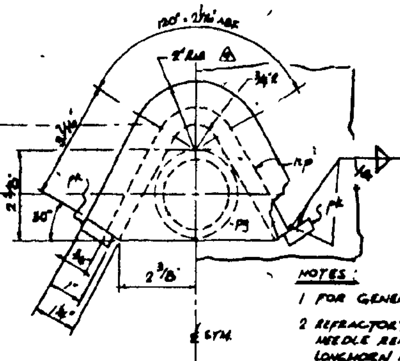


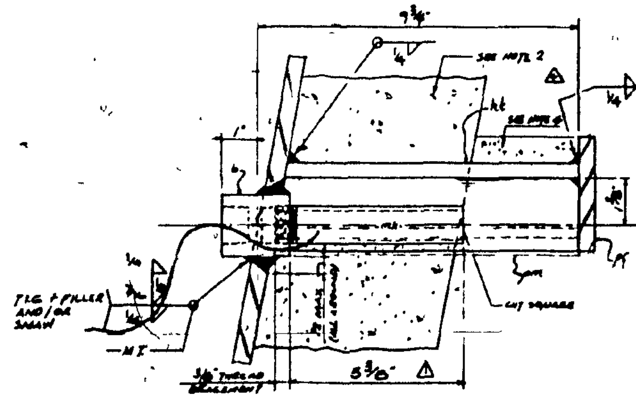
DETAIL-5

MATERIAL FOR ONE (1) WASTE HEAT EXCHANGER				
QTY	MARK	DESCRIPTION	LENGTH	MATERIAL
1	ha	PL 1/2 x 140	24'-6"	SA 516 GR.70
1	hb	(CUT 1-ha & 1-hb		" "
1	hc	PL 1/2 x 3	14'-0 5/8"	" "
1	hd	PL 5/8 x 13 1/4	16'-10"	" "
1	hf	PL 1/2 x 10 1/4	2'-1 1/4"	" "
2	hg	1/2" ϕ ROUND BAR	2 x ...	A 36
3	mb	2" ϕ x 218 WALL (SMLS) PIPE	0'-1 1/16"	SA 106 GR B
1	md	FLANGE RING		SEE ON SK #1
2	mf	1/4" ϕ ROUND BAR	1'-3"	A 36
2	mg	1/4 ϕ ROUND	1'-9"	"
3	mh	2" ϕ x 218 WALL ... PIPE	5 1/8"	S.S. TYPE 410
2	pc	PL 1 5/16 x 8	1'-8"	SA 516 GR 70
1	pd	PL 3/4 x 2 1/2	1'-11"	CSA G40.21 44W
3	a	2" ϕ 300# SLIP-ON FLANGE		SA-105
1	b	3/4" ϕ 6000# SCREWED HALF COUPLING		SA 105
1	d	24" ϕ 150# SLIP-ON FLANGE		" "
1	d	24" ϕ 50# BLIND FLANGE		" "
1	SC	DAVIT		...
1	hk	1/4 ϕ ROUND BAR	5'-2 7/8"	A-36
2	hm	1/4 " "	1'-9 1/2"	
2	hn	1/4" ϕ "	1'-5 1/4"	"
3	hp	PL 1/2 x 8 5/16	9 1/2"	S.S. TYPE 410
1	hc	PL 1/2 x 3 13/16	9 7/8"	
1	pf	PL 1/2 x 3 7/16	5 7/8"	" "
3	pg	PL 1/2 x 4 3/8	6 7/8"	"
1	mn	... x .154 WALL ... PIPE	3 1/16"	S.S. TYPE 410
80	P1	1 1/8" ϕ STUD BOLTS	7 1/4"	...
40	P2	1 1/8" ϕ HEX NUTS		...
1	G1	1/8 ... GASKET		...
6	pk	PL 3/8 x 1	3 1/2"	S.S. TYPE 410
2	pm	PL 3/8 x 1	3 1/16"	" "

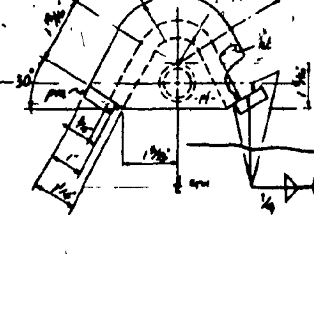


DETAIL-2





DETAIL-4



NOTES:

1 FOR GENERAL NOTES SEE DWG GN-1.

2 REFRACTORY 5 1/2" THK. RESCO RS 17E (BY OTHERS) NEEDLE REINFORCED ON WIGGLY Y STUDS OR LONGHORN ANCHORS AS FOLLOWS:

a) NEEDLES TYPE MELTEX NBR 19-11 ... 1" LONG FULLY ANNEALED, FOR GUNNING, AT A RATE OF 3.5 LBS/100 LBS OF REFRACTORY

b) STUDS, 304 S.S. WIGGLY Y-STUDS OR LONGHORN ANCHORS ON MAX 11 INCHES CENTERS, 10% OF STUDS TO BE THREE PRONGED DEPTH GAUGE STUDS.

NEEDLES AND STUDS ARE SUPPLIED BY REFRACTORY SUPPLIER AND WELDED IN BY D.B.Co.

3 REFRACTORY SUPPLIER TO PACK THIS AREA WITH KAOWOOL BULK A

4 3/4" THK OF AA-22 PLUS NEEDLES APPLIED ON HEXMESH; FOR HEXMESH SEE DWG #2.

STAMPING SPACE

DRAWN: E. MAPLES AUG 21/78

CHECKED: Feb/23/79

APPROVED: July 26/79

A-B

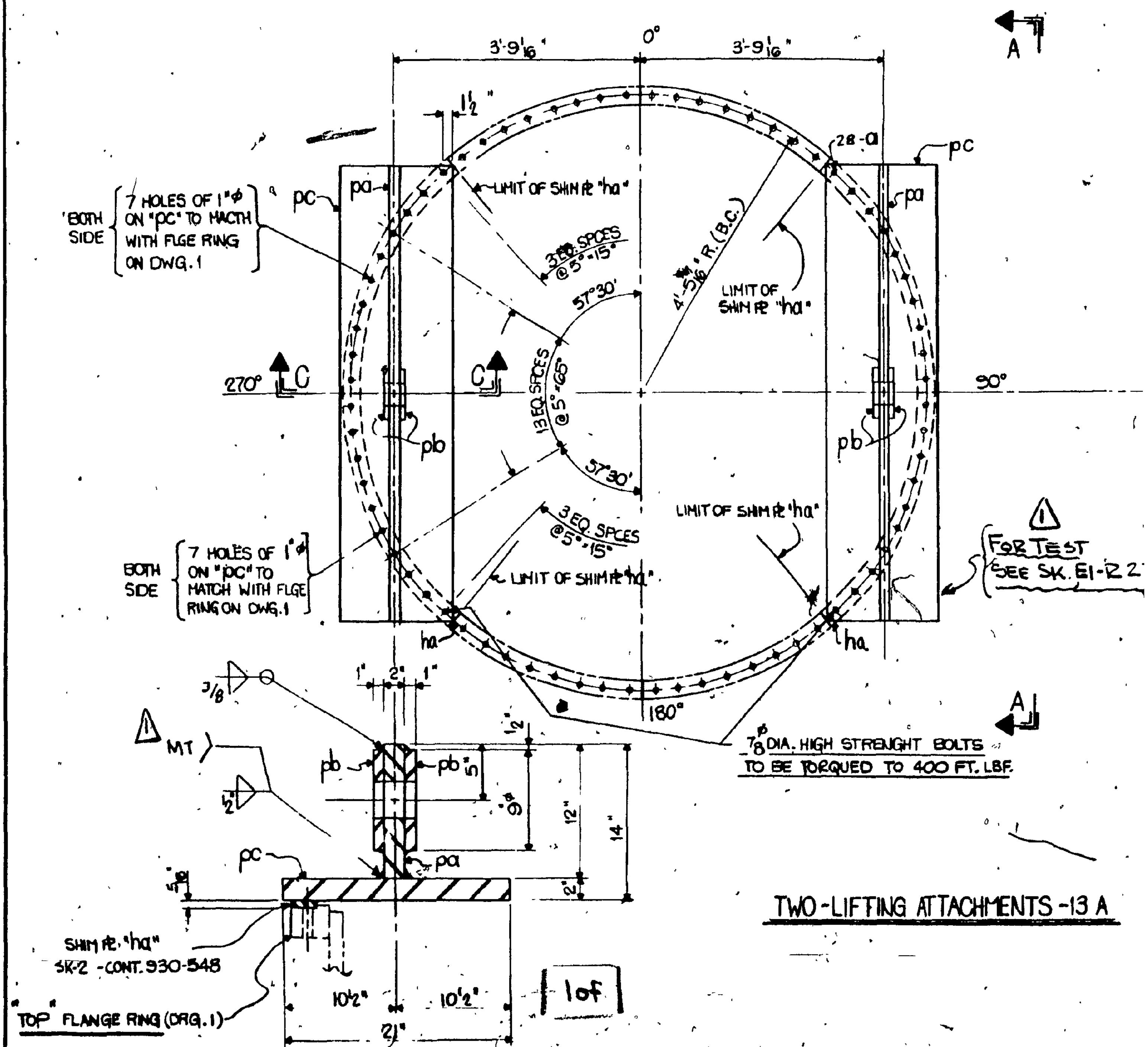
Dominion Bridge Company, Limited

ÉQUIPEMENT SUR COMMANDE / ENGINEERED PRODUCTS

DESCRIPTION: ONE (1) - FEIN WASTE HEAT EXCHANGER

DETAIL: GAS INLET CONE

8



SECTION C-C

TWO-LIFTING ATTACHMENTS -13 A

1 of

No.	REVISIONS		DATE
1	MT FOR WELD PC FREE TO SK.	GD	

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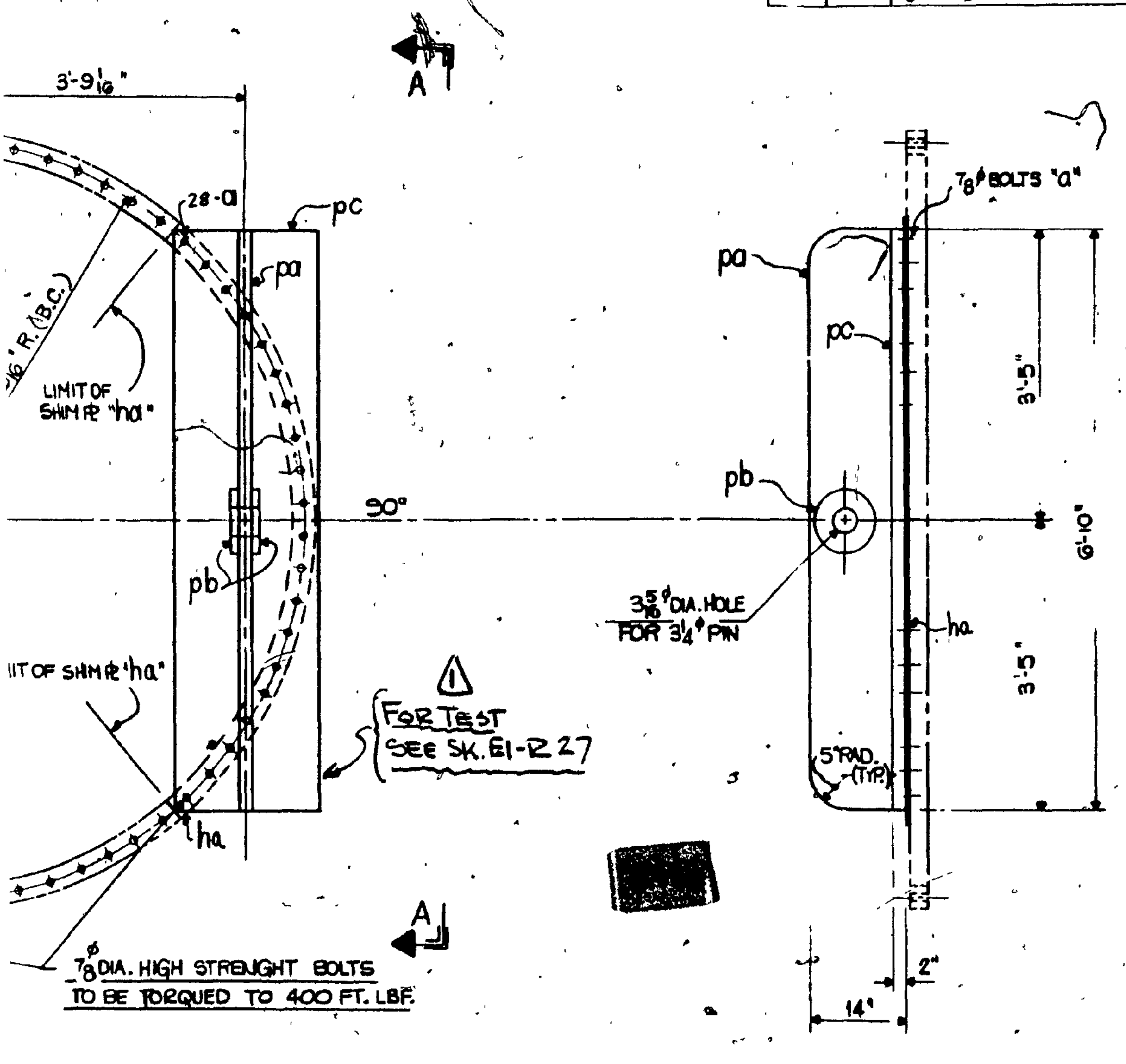
DESSINÉ / DRAWN: G. BOYER	DATE: SEPT/17/79
VÉRIFIÉ / CHKD: GD	DATE: SEPT/18/79
APPROUVÉ / APPROVED: J. Skandale	DATE: 18/9/79
SECTION: BOILER - 493	

FABRIQUÉ AU CANADA / MADE IN CANADA

BORDEREAU DES MATÉRIAUX / MATERIAL BILL: 13A

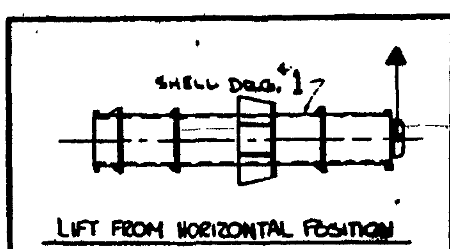
APPROBATION DES SOUDURES / WELDING APPROVAL: Allen Goode

MATERIAL FOR ONE		A			
QTY	MARK	DESCRIPTION	LENGTH	ZONE	MATERIALS
2	pa	℞ 2" x 12"	6'-10"		SA 516 GR 70 OR CSA G40.21 GR 44T
4	pb	℞ 1" x 9"ϕ DIA.			
2	pc	℞ 2" x 21"	6'-10"		
2	ha	℞ 5/16"			SHOWN ON SK-2 CONT. 980-546
28	a	7/8ϕ H.S.B. SA325	7"		FS5620Q



VIEW A-A

TWO-LIFTING ATTACHMENTS -13 A



LIFT FROM HORIZONTAL POSITION

DESSINÉ / DRAWN: G.BOYER	DATE: SEPT/17/79	FABRIQUÉ AU CANADA / MADE IN CANADA	**Dominion Bridge Company, Limited**
VÉRIFIÉ / CHKD: G.W.	DATE: SEPT/18/79	BORDEREAU DES MATÉRIAUX / MATERIAL BILL: 13A	DIVISION DES PRODUITS INDUSTRIELS / INDUSTRIAL PRODUCTS DIVISION
APPROUVÉ / APPROVED: Y. Kandel	DATE: 18/9/79	APPROBATION DES SOUDURES / WELDING APPROVAL: Allen Gourlie	DESCRIPTION: ONE(1) FCIM WASTE HEAT EXCHANGER
SECTION: BOILER - 493			DETAIL: TWO LIFTING ATTACHEMENTS
			PROPRIÉTAIRE / PROPRIETOR:
			CLIENT / CUSTOMER: IMPERIAL OIL LTD.

N° DE VOÛTE / VAULT NO: | DESSIN / DWG.: 13 | REV: 1 | CONT: 565-981

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